From: <Bob.Fetterman@valves.spx.com>
To: "Phong Do" <PHONG-D@ipsc.com>
Date: 8/24/2004 5:47:33 AM
Subject: Re: Parallel Slide Valve Comment

Phong,

It is my personal opinion that a parallel sliding gate valve should be adequate for your application.

Bob

Robert L. Fetterman
SPX VALVES & CONTROLS
DeZURIK * COPES-VULCAN * MUELLER STEAM * DANIEL VALVE * FEBCO * POLYJET * K-FLO
Senior Applications Engineer
Copes-Vulcan
5620 West Road
McKean, PA 16426-1504
814-476-5816 (Voice)
814-476-5854 (FAX)
bob.fetterman@valves.spx.com
www.dezurikcopesvulcan.com

"Phong Do" <PHONG-D@ipsc.com>
08/23/04 04:27 PM

To
<Bob.Fetterman@valves.spx.com>
cc

Subject
Re: Parallel Slide Valve Comment

Bob,
We are not replacing a Copes-Vulcan globe valve. This is a different

project. The existing glove valve uses air actuator. There is no stroke time requirement. There is no other valve in the loop. This valve may be opened/closed 3-4 times per year.
PTD.

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 9/1/2004 7:55:58 AM
Subject: Re: IP Clearance Data

Hi Phong,

I see nothing wrong with your clearance data. We also believe that the problem may be due to steam leakage past the first stage diaphragm on the steam end flow.

Attached is my preliminary calculation report on the orifice sizing and system installation. Would you please review to see if this satisfies your immediate needs.

One thing I am not clear about is your standard mode of operation - your guidance would be useful.

I assume that for cold starts HSPV is used to pressurise the HP casing and warm it through by condensation heating. This would normally take place whilst condenser vacuum is being raised, at which point the IP rotor will also be heated through HSFV from the gland packing lines.

Once the turbine is ready to roll, HSPV will close and the HP cylinder will depressurise.

HSFV will then close and S11 will open.

If I am correct on the above assumption then I foresee no problem. If, however, S11 is programmed to open before the HP cylinder depressurises then there may be a drainage problem and we may need to consider a drain orifice to bypass the new pneumatic isolating valve.

Note also, do you believe it would be necessary to supply this valve with limit switches? These could be arranged in parallel to the limit switches on S11 to signify correct operation.

An estimate of the performance benefit and the mark-up to 903E892 will follow.

Regards,

Alan

(See attached file: IPS0001A2.pdf)

CC: <kevin.spires@power.alstom.com>, <robert.cunningham@power.alstom.com>

From: <kevin.spires@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 7/15/2004 6:55:28 AM
Subject: Re: IPSC IP Cooling Temperature Reduction

Hi Phong

Sorry for delay in getting back to you, but I have been in Sweden yesterday and had problem getting mail.

ALSTOM will be happy to get involved in this and I believe the best way forward is that you look at purchasing the 1" pipe, the valves and connections to the other pipes and do the labour from within IPSC or a local company.

If you could send me an estimate of what it will cost in terms of labour and materials, I can then look at raising an order on you for carrying out the work in March 05.

Based on previous correspondence, I think you have all the details you need to give me your best guess but if you want further clarifications on pipe and/or materials please do not hesitate to contact me.

Regards

Kevin

Phong Do <PHONG-D@ipsc.com> on 14/07/2004 14:19:07

To: Kevin SPIRES/GBRUG01/Power/ALSTOM@GA, Robert BROWN/GBRUG01/Power/ALSTOM@GA, Robert CUNNINGHAM/GBRUG01/Power/ALSTOM@GA
cc: James Nelson <JIM-N@ipsc.com>, Alan HOLMES/GBRUG01/Power/ALSTOM@GA

Subject: IPSC IP Cooling Temperature Reduction

Kevin,
Per our previous discussion, we wondering if Alstom would consider the following:
(a) Provide cost reimbursement for the U1 IP cooling steam temperature reduction project, including design, material and installation labor, or
(b) Take a direct lead, with IPSC directions, to design, supply parts and to install of the project in the upcoming March 2005 U1 outage.

Recent data verifies that (1) the U1 pos- refurbishment IP cooling temperature has higher enthalpy/temperature comparing to the U1's pre-refurbishment and also higher than the U2's post-refurbishment, immediately after the HP retrofit (2) the higher IP cooling temperature would reduce the service life of the IP rotor, therefore warrant the implementation of the temperature reduction project.

We appreciate Alstom for the effort of resolving the above problem by provided a design concept. However, since the new U1 HP retrofit contributes to the higher IP cooling steam temperature, we suggest Alstom to consider the above listed "a" or "b" option.

It was a great experience to work closely with Alstom to realize its product benefit. I looking forward to hearing from you. Please response by July 23, 2004.

Phong Do
Day Phone: (435)864-6475
Fax: (435)864-0775
Email: Phong-D@IPSC.Com

>> U2IPCoolingTempMeasuring,6,2,04.xls.xls removed by Kevin SPIRES on 15 July 2004
>> U1IPCoolingTempMeasuring,6,2,04.xls.xls removed by Kevin SPIRES on 15 July 2004

:.________________

CC: <robert.brown@power.alstom.com>, <robert.cunningham@power.alstom.com>, James Nelson <JIM-N@ipsc.com>, <alan.holmes@power.alstom.com>

	Intermountain Unit 2	Intermountain Unit 1
Main Steam Pressure / psia	2412.2	2412.2
Main Steam Temperature / °F	1000	1000
After HP stage 1 Pressure / psia	2010.1	1995.8
After HP stage 1 Temperature / °F	948.6	946.5
After HP stage 2 Pressure / psia	1739.7	1728.9
After HP stage 2 Temperature / °F	904.5	902.7

Supplied by Alstom on Jan 2004

From: James Nelson
To: Phong Do
Date: 7/27/2004 11:53:45 AM
Subject: Fwd: Intermountain Power Service

>>> <lucy.liu@power.alstom.com> 7/27/2004 9:18:57 AM >>>

James Nelson,

It was nice to talk to you yesterday.

Based on our conversation, by my understanding, you may have two problems:

(1) Unit 1 may have a higher HP exhaust temperature, which drives higher IP bowl cooling temperature. You think throttle condition is normal.

(2) Unit 2's main steam control valves position is 9% more closed than Unit 1. You think the HP 1st and 2nd stages flow path area is bigger than you required.

Here are my questions and concerns:

1. Could you please explain me more about the background that why you wanted to limit the flow before the retrofit?

2. On Unit 2 Valve Position Issue:

   I think both Unit 1 and Unit 2 should be the same design. When did you find Unit 2's control valve position was 9% more closed than Unit 1? May I have the acceptance test (Performance Test) data?

   Valve 9% more closed means valve is only 91% opened and there is only 91 % of designed steam flow passing through HP turbine. Reduced flow (roughly 9%) will cause approximately 9% of output losses which is about 78MWs. Do you lost these MWs? It doesn't look like you lost so much output. I doubt the valve position calibration or reading or other instrumental problems. Could you please check and confirm valve's problem?

   If the position is correct, there are two things imaginable: (1) the valves that on the lines bypassing HP may be opened; (2) HP rear deposition or any other reason that reduced HP flow path area.

   Since I don't have Heat Balance and P&I diagram right now, I just imagine that you may have a main steam bypass line, which bypass HP. This line mainly is used for start up and shut down. You need to make sure the valve on this line is closed during the operation. Additionally please do a valve check on: the valve spindle leakage, the valve on the line from the point before MSV to BFW turbine inlet, and all the other valves.

Only if we exclude all these possible problems then we may start thinking about the HP flow area change problem. Since you don't have tag to measure HP loop pressure, in this case, a performance test (enthalpy test) is needed. I suggest you clear all these possible problems first for Unit 2 before the test.

3. On Unit 1 HP exhaust temperature

I order to identify this problem; we need the operation data for analysis. As the first step we only analyze the HP area. Please provide the following information:

(1) Heat Balance diagrams;
(2) IP Section drawing;
(3) P&I diagrams
(4) Data in the attached sheet (hour average; also please include the definition of each tag)

(See attached file: Intermountain Unit_1 HP data requirment.xls).

Please contact to me any time when you need.

Best regards,

Lucy Liu

Intermontain Unit 1 HP Performar

PI Data Start Time
PI Data End Time
Tag number and its parameter that the tage stands

Measured load
Corrected load

HP Cylinder Efficiency
Standard Cylinder HP Efficiency
Deviation from standard HP Cylinder

IP Cylinder Efficiency
Standard Cylinder IP Efficiency
Deviation from standard IP Cylinder

HP Valves & Cylinder Efficiency
Standard HP Valves & Cylinder Efficiency
Deviation from standard Vlvs & Cyl

IP Valves & Cylinder Efficiency
Standard IP Valves & Cylinder Efficiency
Deviation from standard Vlvs & Cyl

LP Efficiency

Pressure

Main steam pressure
First(Loop) Stage pressure
Cold reheat pressure
Hot reheat pressure
#1 Htr extr pressure
#2 Htr extraction pressure
#3 Htr extraction pressure
Crossover pressure
DA Selected pressure
#5 Htr extrl pressure
#6 Htr extr pressure
#7Htr extr pressure
#8 Htr extr pressure
Backpressure

Temperatures
Main steam temperature
First stage temperature
Cold reheat temperature
Hot reheat temperature
#1 Extraction temperature
#2 Extraction temperature
3 Heater extraction temp

4 Heater extraction temp
5 Heater extraction temp
6 Heater extraction temp
#7 Heater extraction temp
#8 Heater extraction temp
Crossover Temp
Hot well temperature

#1Hr. Feedwater inlet temp
#2Hr. Feedwater inlet temp
#3 Feedwater inlet temp
#DA condensate inlet temp
HPBFP suction temp
#5 condensate inlet temp
#6 condensate inlet temp
#7 condensate inlet temp
#8 condensate inlet temp

Economizer Inlet temp
Condenser cooling water temperature

#1 Hr. drain temp.
#2 Hr. drain temp.
#3 Hr. drain temp.
#5 Hr. drain temp.
#6 Hr. drain temp.
7Hr. drain temp.

#8 Hr. drain temp.

BFP turbines information
Flow
Extraction pressure and temperature
Exhaust pressure and temperature

Bearing information includes
Bearing temperature
Bearing vibration and thrust bearing position

Flows
SH Spray Flow
RH Spray Flow
Final Feedwater
Condensate flow
Condenser cooling water flow

Other
TEMP DROP ACROSS RH ATTEMP
REHEATER PRESSURE DROP
FIRST STAGE TO #1 EXTR PRESSURE RATIO
FIRST STAGE TO HP EXHAUST PRESSURE RAT
#1 EXTR TO HP EXHAUST PRESSURE RATIO
IP INLET TO EXHAUST PRESSURE RATIO
FIRST STAGE PRESS TO THROTTLE FLOW RAT
HP HTR DRAIN FLOW TO FW FLOW RATIO
CONDENSATE TO FEEDWATER FLOW RATIO

ıce Analysis Data Requirement

for

IP7008438

10

10

From: Phong Do
To: Jim Knapp
Date: 8/18/2004 7:10:26 AM
Subject: New S11 Valve for the New Line

Jim,
In an effort of reducing the U1 IP cooling steam temperature, we are considering:
1. Install new line which runs from HP cold reheat to the existing IP cooling line to mix and cool the steam.
2. Install new S11 and SRCV (ACV-19 and ACV-20, 1TGC-M4080A) on the new line for performance and overspeed protection.

Please let me know if the existing S11 (ACV-19) has a single acting or double acting actuator as this will affect the pneumatic interface with the new valve. It would be good if you can send me the spec of this actuator and solenoids.

I will go over with you when the design package is completed. Thank you much.

>>> <alan.lang@power.alstom.com> 8/18/2004 1:30:59 AM >>>
Hello Phong,

We agree to the need for an isolating valve in the new cooling line since steam from the pressurised HP cylinder will be less effective at warming the IP rotor than steam from the gland sealing line.

We propose to use a parallel slide gate valve as this will require less operating force than a globe valve - is this OK?

Would you also find out if the existing S11 has a single acting or double acting actuator as this will affect the pneumatic interface with the new valve.

Thank you and best regards,

Alan

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GE Energy

INTERMOUNTAIN POWER UNIT 1 IP ROTOR COOLING STEAM ANALYSIS AND REPORT

for

Intermountain Power Service Corporation
Unit 1

Equipment Serial #: 270T150

June 9, 2004

Cecil James
West Region Applications Engineer

Bob Fink
Service Manager

Matt House
Account Manager

Bill Kuehn
Field Marketing Director

EXECUTIVE SUMMARY:

Intermountain Power Service Corporation (IPSC) asked GE Energy to help resolve the current high temperature of the Unit 1 (250T150) intermediate pressure (IP) rotor cooling steam. This problem with elevated temperatures was immediately subsequent to a turbine outage during which Alstom installed a dense pack rotor and diaphragms.

GE's design for cooling the IP rotor utilizes 2nd stage steam extracted from the HP, throttling the steam to a determined pressure and admitting it to the IP rotor near the 8th stage diaphragms. Maintaining proper cooling steam temperature is important to realize the IP rotor's full useful life. Temperatures above those specified by GE will effectively reduce the life of the rotors and result in additional costs due to ancillary rotor inspections and premature replacement of the IP rotors.

GE strongly discourages raising any IP cooling steam alarm set points. Instead, GE encourages a prompt resolution to the root cause.

Data indicates the fit between the Alstom inner shell and the extraction snout may be leaking and is allowing 1st stage steam to leak into the cooling extraction to the IP rotor.

GE Energy will support IPSC to quickly resolve this issue but feels it would be in IPSC's best interest if Alstom first tries to resolve any extraction snout fit problems as part of the refurbishment warranties and before having to incur significant modifications from re-routing appropriate steam to the IP turbine rotor. If for any reason Alstom is not able to perform to IPSC's expectations, GE will proceed with supporting a quick resolution to IPSC's satisfaction.

Until a long-term resolution is implemented we recommend the following inspections during the next reasonable maintenance outage:

1. Inspect the IP rotor cooling steam flow-restricting orifice (148D5583) and replace as needed.
2. Remove the HP 2nd stage extraction snout and inspect for steam cutting, mechanical damage, cracking, etc. and replace as needed. IPSC's preferred method for NDE may be necessary to identify all defects.
3. Inspect the extraction snout to inner shell fit for steam cutting, mechanical damage, cracking, etc. A borescope, with remote video, would facilitate documenting this suspected area for a more thorough review.
4. Inspect and calibrate thermocouples T-1 and T-2 (148D5583).

At IPSC's request, GE Field Services can assist these inspections and further identify the causes for this IP cooling steam temperature problem.

In the interest of realizing the IP rotor's full life, GE recommends a resolution be implemented as soon as IPSC's operation permits.

Introduction:
Intermountain Power Service Corporation (IPSC) asked GE Energy to assist resolving the high temperature of the cooling steam to the Unit 1 (270T150) intermediate pressure (IP) turbine rotor. This problem of elevated temperatures was immediately subsequent to a turbine outage during which Alstom installed a dense pack rotor and diaphragms. Intermountain Power Unit 2 (270T151) was also retrofitted with an Alstom dense pack but does not have the same IP rotor cooling temperature problems as Unit 1.

GE's design for cooling the IP rotor utilizes 2nd stage steam by extracting the steam from the HP immediately downstream from the stage two buckets, throttling the steam to a determined pressure and admitting it to the IP at the 8th stage diaphragms. IPSC attempted to reduce cooling steam temperature by adjusting the throttling valve, but realized only marginal results.

GE's Six Sigma ACFC methodology will be implemented to analyze the data as a means of identifying factors that may be causing this problem and identify a reasonable remedy.

Define (Problem Statement):
The 8th stage turbine end temperature is currently indicating 934 degrees F. Normal temperature for this location is 917 degrees F and alarm is 937 degrees F.

Measure:
Phong Do, IPSC Mechanical Lead Engineer, reported the following data taken from IPSC's acceptance testing:

IPSC's STEAM TURBINE HP TO IP ROTOR COOLING STEAM DATA			
Updated on 5/2/03			
Data	Unit 1	Unit 2	Comments
Generator Power Output, MW	950	982	
IP Cooling Steam Static Pressure, Psia	475.7	512.12	
Pre-retrofit IP Cooling Steam (Temperature°F, Flow Rate lb/hr)	(828, 12372) Note (5)	Not Available	
Post-retrofit IP Inlet Steam Temperature, °F	861. Note (3)	843.5. Note (4)	
Post-retrofit IP Inlet Steam Enthalpy, BTU/lbm	1446.5	1435.7	Appended per C. James, GE for calculation purposes
Differential Pressure, inches of H_2O	1460.4	649.6	
IP Cooling Steam Mass Flow Rate, lbm/hr	26587.6	18987.4	
Post-retrofit IP Inlet Rotor Turbine End Temp, °F	934	915	
Post-retrofit IP Inlet Rotor Generator End Temp, °F	913	899	
Pre-retrofit IP Inlet Rotor Turbine End Temp, °F	918	912	
Pre-retrofit IP Inlet Rotor Generator End Temp, °F	900	902	

Table 1: 270T150, 270T151 Acceptance Data

IPSC performed additional testing to further identify and validate current operating conditions (actual point of TC contact was not validated for this analysis):

IPP
INTERMOUNTAIN POWER SERVICE CORPORATION

IP Cooling Temperature

Unit: U1

Date: 2-Jun-04 | Prepared by: Jim Young

Time Stamp	Time Duration (hrs)	IP Cooling Temp (F)	Generating MW Output (MW)	8th Stage, TE Temp (F)	8th Stage, GE Temp (F)	Main Steam Pressure (psig)	Main Steam Temp (F)	Hot Reheat Pressure (psig)	Hot Reheat Temp (F)
	H	C	D	E	F	G	H	I	J
				1TGATE-0080	1TGATE-0081	1SGGPT-0001	1SGGTE-0004	1SGJPT-0006	1SGJTE-0013
6/2/04 7:41 AM		810	916	934	913	2410	1010	532	996
6/2/04 8:32 AM	0:51	808	953	921	896	2357	1000	556	989
6/2/04 9:30 AM	1:49	812	954	938	908	2400	1007	556	1001
6/2/04 10:39 AM	2:58	807	955	938	911	2395	1000	559	998
6/2/04 11:33 AM	3:52	814	957	933	913	2411	1012	556	1011
6/2/04 12:48 PM	5:07	806	954	930	903	2391	1003	558	992
Avg		810	948	932	907	2394	1005	553	998

Table 2: 270T150 Field Data

IPP
INTERMOUNTAIN POWER SERVICE CORPORATION

IP Cooling Temperature

Unit: U2

Date: 2-Jun-04 | Prepared by: Jim Young

Time Stamp	Time Duration (hrs)	IP Cooling Temp (F)	Generating MW Output (MW)	8th Stage, TE Temp (F)	8th Stage, GE Temp (F)	Main Steam Pressure (psig)	Main Steam Temp (F)	Hot Reheat Pressure (psig)	Hot Reheat Temp (F)
	H	C	D	E	F	G	H	I	J
				2TGATE-0080	2TGATE-0081	2SGGPT-0001	2SGGTE-0004	2SGJPT-0006	2SGJTE-0013
6/2/04 7:48 AM		786	951	918	911	2364	1009	559	1010
6/2/04 8:36 AM	0:48	777	950	912	894	2354	1001	560	993
6/2/04 9:35 AM	1:47	774	948	914	896	2374	1001	561	1001
6/2/04 10:43 AM	2:55	782	947	914	895	2378	1001	560	996
6/2/04 11:40 AM	3:52	779	947	914	896	2386	1002	562	1005
6/2/04 12:53 PM	5:05	784	950	914	896	2373	1002	564	994
Avg		780	949	914	898	2372	1003	561	1000

Table 3: 270T151 Field Data

A gage R&R analysis is not available based on the limitation of a single operator taking the data with a single instrument.

Analyze:

Table 1 indicates a significant increase in Unit 1 steam temperature and flow:

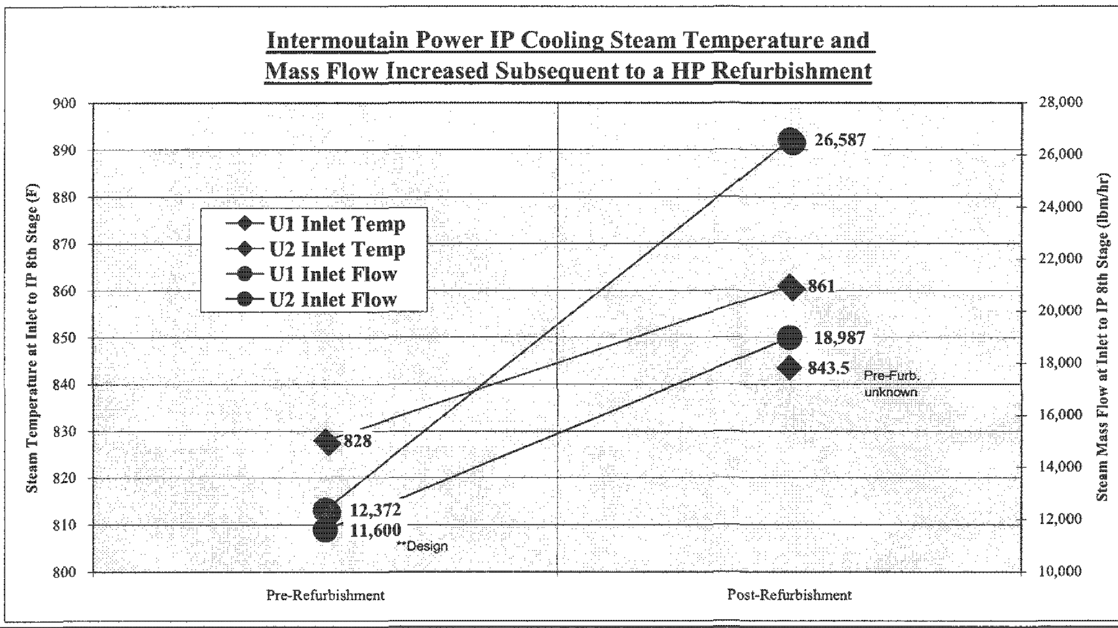


Figure 2: Steam temperatures and flow to the U1 IP have increased significantly

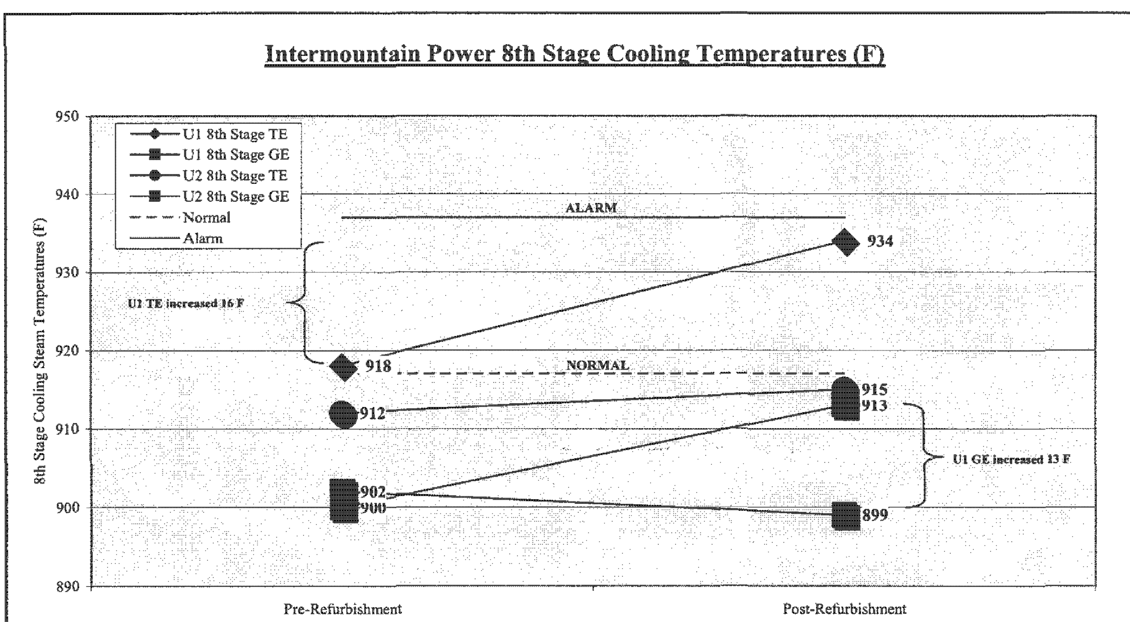
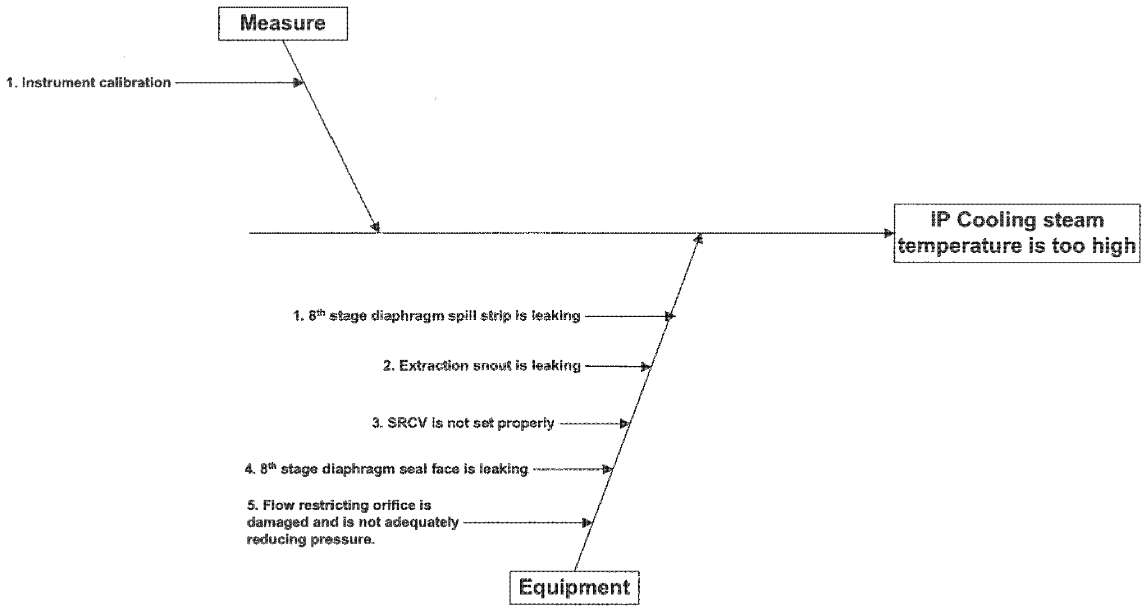


Figure 1: Unit 1 IP TE and GE rotor cooling temperatures increased 16F and 13F respectively.

IP7008447

A cause and effect analysis helped identify possible causes to the problem of higher steam temperatures:



Measure:
1. The TCs were calibrated and verified – discounting this as an issue.

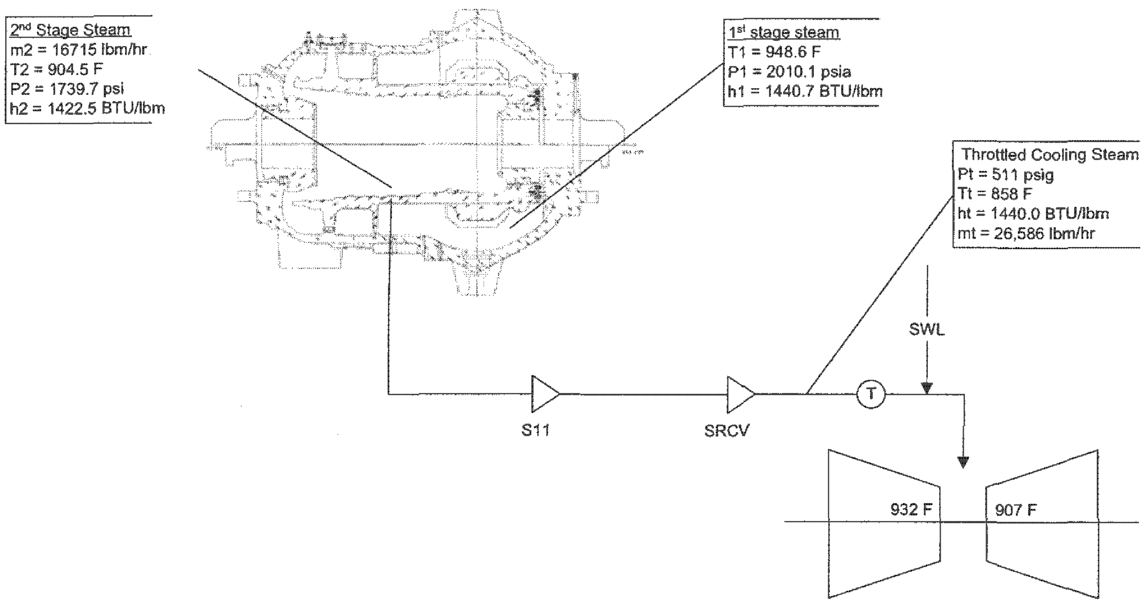
Equipment:

1. No known QA documentation for the as-left 8th stage spill strip. Since this data is readily taken and verified during the assembly process, the information would have to be reviewed and confirmed.
2. No known QA documentation for installing the 2nd stage extraction snout.
3. Phong Do reported an effort to optimize steam flow and temperature based on SRCV position. No significant improvement was realized by turning down the SRCV. The SRCV was left at the point minimum temperature was measured.
4. The 8th stage diaphragm was rebuilt during the outage at the Salt Lake GE I&RS service center. QA documentation was verified and does not indicate any significant discrepancies that could lead to the elevated steam temperatures, and since elevated temperatures are measured prior to the IP inlet any diaphragm leakage would be only supplementary and not the major contributor to this problem.
5. SRCV optimization data performed by IPSC measured an average static pressure of 511 psig at the flow measuring orifice which is downstream of the pressure reducing orifice and appears to be operating acceptably.

The most significant issue is the mass flow rate measured at the Unit 1 flow-measuring orifice. Phong Do verified the 2.00" orifice and confirmed the orifice in good condition and it's dP values acceptable. The measured cooling steam flow of 26,587 lbm/hr is approximately 9,872 lbm/hr over the design of 16,715 lbm/hr.

A heat balance of the extraction system will be used to identify the source of the extra flow:

UNIT 1

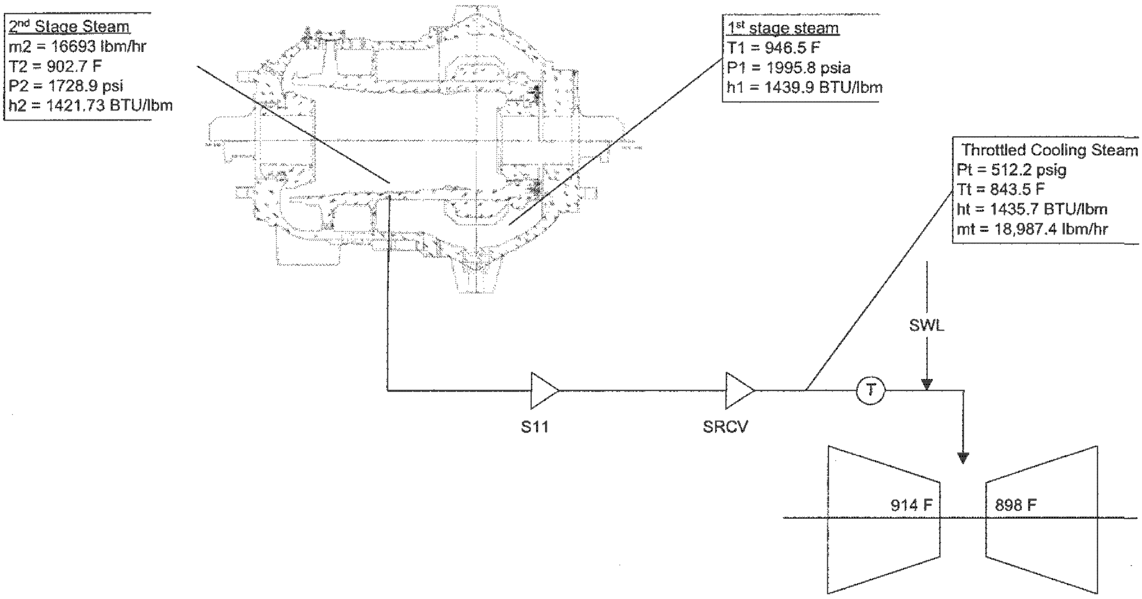


UNIT 1 HEAT BALANCE			subscripts:	*t = total IP cooling flow*	*1 = supplementary*	*2 = 2nd stage flow*
mtht = m1h1 + m2h2			Color Code:	**Design Data**	**Measured Data**	**Calculated Data**
P1 =	2010.1	psia		1st stage pressure per IPSC		
T1 =	948.6	F		1st stage temperature per IPSC		
Cal. H1 =	1440.7	BTU/lbm		1st stage enthalpy		
m1 =	9,871.8	lbm/hr		supplementary cooling steam flow to IP (mt - m2)		
h1 =	**1,480.4**	BTU/lbm		Enthalpy of supplementary steam = (mtht - m2h2)/m1		
m1 * h1 =	14,614,374.9	BTU/hr		Supplementary steam energy = mtht - m2h2		
P2 =	1,739.7	psia		2nd stage pressure per IPSC		
T2 =	904.5	F		2nd stage temperature per IPSC		
m2 =	16,715.0	lbm/hr		Cooling flow per IPSC		
h2 =	1422.46	BTU/lbm		2nd Stage enthalpy		
m2 * h2 =	23,776,469.6	BTU/hr		2nd stage steam energy to IP (m2*h2)		
Pt =	511.2	psia		Throttled pressure, mean from SRCV Optimization		
Tt =	858.8	F		Throttled temperature, mean from SRCV Optimization		
mt =	26,586.8	lbm/hr		Throttle mass flow, mean from SRCV Optimization		
ht =	1,444.0	BTU/lbm		Throttled enthalpy, mean from SRCV Optimization		
mt*ht =	38,390,844.4	BTU/hr		Cooling steam energy to IP (mt*ht)		

IP7008449

A similar heat balance for Unit 2 is also used to identify any leakage sources.

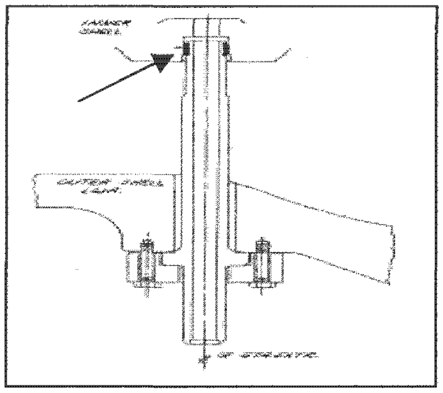
UNIT 2



UNIT 2 HEAT BALANCE			subscripts: *t = total IP cooling flow*	*1 = supplementary*	*2 = 2nd stage flow*
mtht = m1h1 + m2h2		Color Code:	**Design Data**	**Measured Data**	**Calculated Data**
P1 =	1995.8	psia	1st stage pressure per IPSC		
T1 =	946.5	F	1st stage temperature per IPSC		
Cal. H1 =	1439.9	BTU/lbm	1st stage enthalpy		
m1 =	2,294.4	lbm/hr	supplementary cooling steam flow to IP (mt - m2)		
h1 =	1,537.3	BTU/lbm	Enthalpy of supplementary steam = (mtht - m2h2)/m1		
m1 * h1 =	3,527,114.9	BTU/hr	Supplementary steam energy = mtht - m2h2		
P2 =	1,728.9	psia	2nd stage pressure per IPSC		
T2 =	902.7	F	2nd stage temperature per IPSC		
m2 =	16,693.0	lbm/hr	Cooling flow per IPSC		
h2 =	1421.73	BTU/lbm	2nd Stage enthalpy		
m2 * h2 =	23,733,011.8	BTU/hr	2nd stage steam energy to IP (m2*h2)		
Pt =	512.2	psia	per IPSC acceptance testing		
Tt =	843.5	F	per IPSC acceptance testing		
mt =	18,987.4	lbm/hr	per IPSC acceptance testing		
ht =	1,435.7	BTU/lbm	Throttled steam enthalpy		
mt*ht =	27,260,126.7	BTU/hr	Cooling steam energy to IP (mt*ht)		

Based on these heat balances, steam having an enthalpy greater than the HP 2nd stage extraction is leaking into the IP rotor cooling steam. The most likely source of steam that would qualify is the steam in the space between the inner and outer HP shells. This steam is essentially 1st stage steam with an enthalpy of approximately 1440 BTU/lbm.

The leak path could be the fit between the inner shell and the extraction snout. If the snout is mechanically damaged or if the inner shell fit is defective, higher energy 1st stage steam could leak into the cooling extraction path and result in a higher cooling steam temperature.



CONCLUSIONS:

GE Energy will support IPSC to quickly resolve this issue but feels it would be in IPSC's best interest if Alstom first tries to resolve any extraction snout problems as part of the refurbishment warranties and before having to incur significant modifications from re-routing appropriate steam to the IP turbine. If for any reason Alstom is not able to perform to IPSC's expectations, GE will proceed with supporting a quick resolution to IPSC's satisfaction.

Until a long-term resolution is implemented we recommend the following inspections during the next reasonable maintenance outage:

	Inspection	Why	How
1.	Inspect the IP rotor cooling steam flow-restricting orifice (148D5583) and replace as needed.	Assure pressure reducing is as design.	Remove orifice from between flanges and inspect for erosion, damage, etc.
2.	Remove the HP 2nd stage extraction snout and inspect for steam cutting, mechanical damage, cracking, etc. and replace as needed. IPSC's preferred method for NDE may be necessary to identify all defects	Assure the extraction snout is not damaged.	Remove extraction snout and inspect. Replace the extraction snout using a new gasket.
3.	Inspect the extraction snout to inner shell fit for steam cutting, mechanical damage, cracking, etc. A bore-scope, with remote video, would facilitate documenting this suspected area for a more thorough review.	Assure the inner shell fit is not damaged.	Use a bore-scope to remotely inspect inner shell fit and document findings.
4.	Inspect and calibrate thermocouples T-1 and T-2 (148D5583).	Assure the thermocouples are functioning properly.	Reference GE instruction manuals.

At IPSC's request, GE Field Services can assist these inspections and further identify the causes for this IP cooling steam temperature problem.

In the interest of realizing the IP rotor's full life, GE recommends a resolution be implemented as soon as IPSC's operation permits.

From: Phong Do
To: cecil.james@ps.ge.com
Date: 1/29/2004 8:07:37 AM
Subject: Re: IP Cooling Resolution

Cecil,
Please see the below.
Due to the complication of U1 outage comparing to U2s (IP inner shell to diaphragm seal face, diaphragm remachining, diaphragm dishing, diaphragm horizontal join remachining and grooving and the 8th TE upper to lower seal face), I'm suspecting that the higher IP steam cooling is the effect/result. At the end of the day, we do need a good fix. Thank you much for your help, Cecil.

1/ *Drawing 298A1699 is not a replacement drawing. This drawing goes back to the original manufacture of the unit. It documents the fit machining dimensions for the diaphragm based on the shell machining.* I don't have this drawing. Can I have a copy to see what we are talking about?

2/ *Do you know where the replacement diaphragm came from and when it was ordered? I am unable to find any record. Phong, a date or PO# may help on our end to see where this diaphragm came from.* We had an 8th stage spare from GE, which was purchased with the machine. In 1996, we installed this spare in U1. We sent the U1 original 8th stage to MD&A to be refurbished. This refurbished diaphragm was then installed in the U2 in 2002. MD&A refurbished the original U2 diaphragm with the same work scope. We then installed it in the U1 in 2003. This is what I remember. I need to confirm this info with Rick Houston. He is taking sick leave for couple months.

3/ *What is the repair pedigree of this replacement diaphragm (types of repairs, who performed the repairs, area check data, etc)?*
See the attached QA guidelines

4/ *Of utmost importance, when the diaphragm was installed at the past outage (not under GE guidance) were the seal face-to-seal face dimensions in the inner shell taken to insure the diaphragm would be assembled correctly.*
During the U1 2003 outage, all IP diaphragm's seal face to inner shell were measured, checked and found to be satisfied. However, when the diaphragm was bolt together, there was a gap at the diaphragm horizontal join. This condition was found on the IP 8th, 9th and 10th diaphragms. The diaphragms were sent to the machine shop (GE in Salt Lake?) to correct (machine off to square) the horizontal join. Also a groove was machined on the diaphragm seal face and a seal bar was machined and installed in the groove.

We didn't have to do this on 2002 U2 outage.

5/ *There is a seal face on the TE of the diaphragm, that if not assembled correctly (refer to 298A1699) could leak high temperature steam to the area of the tc and possibly account for an increased temperature. Installation records form the past outage should be reviewed.*
This seal face was checked by MD&A at the shop (I was told) and was not recheck at installation. Outage report was reviewed and found no significant concerns.

6/ What are the clearances at the root radial; spill strip from the past outage.
All IP diaphragm root radial, spill strip and interstage clearances were within GE specifications.

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 9/15/2004 3:04:25 AM
Subject: Re: IP Clearance Data

Hi Phong,

The information you ask for is given in sections 4.1 and 4.2.

During normal operation, max flow will be 13045 plus 4148 = 17193 lb/hr. Following overspeed the flow will be 5630 plus 6822 = 12452 lb/hr (i.e. marginally less than the anticipated flow you have at present). These flows are calculated on the basis of the reg valves (S11, NEWS11) being fully open and, in the case of overspeed, the HP cylinder having rapidly depressurised to cold reheat conditions.

I agree that the conclusion does not say much at the moment. I am waiting for some thermo data that will give a good estimate of the performance benefits associated with the change. This information will be included in the next issue. At this point in time I would like to be sure that I have fufilled your requirements.

Best regards,

Alan

:.____________________

CC: <kevin.spires@power.alstom.com>, <robert.cunningham@power.alstom.com>

A P & F CONSTRUCTION
1230 WEST 2600 SOUTH
WOODS CROSS, UTAH 84087

Intermountain Power Service Corp.
850 West Brush Wellman Road
Delta, Utah 84624-9546

Sept 29, 2004

Attention: Phong Do

Reference: Steam Line Modifications

Dear Mr. Do;

A P & F Construction is pleased to furnish the following pricing to supply and install as per the walk thru, specifications and drawings that were sent to us. We are assuming that this project will go during the Outage in the spring, if this is the case these prices are probably a little high. Our price for this is as follows:

Labor	$ 5,976.00
Insulation	$ 3,900.00
Materials	$ 2,856.00
Scaffolding	$ 2,900.00
Equipment	$ 900.00
	========
Total	$16,532.00

We are quoting on a ball valve not a gate valve. Price for the isolation valve is $2,025.40. Delivery time is 6 weeks on the isolation valve. We have added 20% to this quotation.

If there are any questions or clarifications on this quotation please feel free to contact me at (801)397-2763x103 or by fax at 397-1899. My e-mail address is apf@att.net

Yours Truly

Ron Pinarelli

From: <robert.cunningham@power.alstom.com>
To: <phong-d@ipsc.com>
Date: 9/28/2004 7:31:43 AM

Rob,
Reference our discussion this morning (following Alans' email dated 25/8/2004).
The effects of changing the IP rotor cooling steam

from

26587 lb/h from after HP stage 2

to

9625 lb/h from after stage + 2875 lb/h from HP exhaust

is Heat rate improvement of approximately 3 Btu/kWh (0.04%)

Regards,
Bob

CC: <robert.brown@power.alstom.com>, <phill.kearney@power.alstom.com>, <RUGWW.TEGMail@test.alstom.com>, <tim.shurrock@power.alstom.com>, <kevin.spires@power.alstom.com>

Intermountain

IP Rotor Cooling Augmentation Calculation

CONTENTS

1. Introduction

2. Modelling The System

3. Derivation of Results

4. Calculation Results

5. Conclusion

Appendix

Date: 23.8.04

1.0 **Introduction**

Following retrofit of the HP cylinder on Intermountain unit 1, excessively high temperatures have been observed at the first stage wheel of the IP cylinder. This is because of high enthalpy steam from the HP second stage tip seals being introduced into the leak-off cooling flow to the IP rotor cooling system.

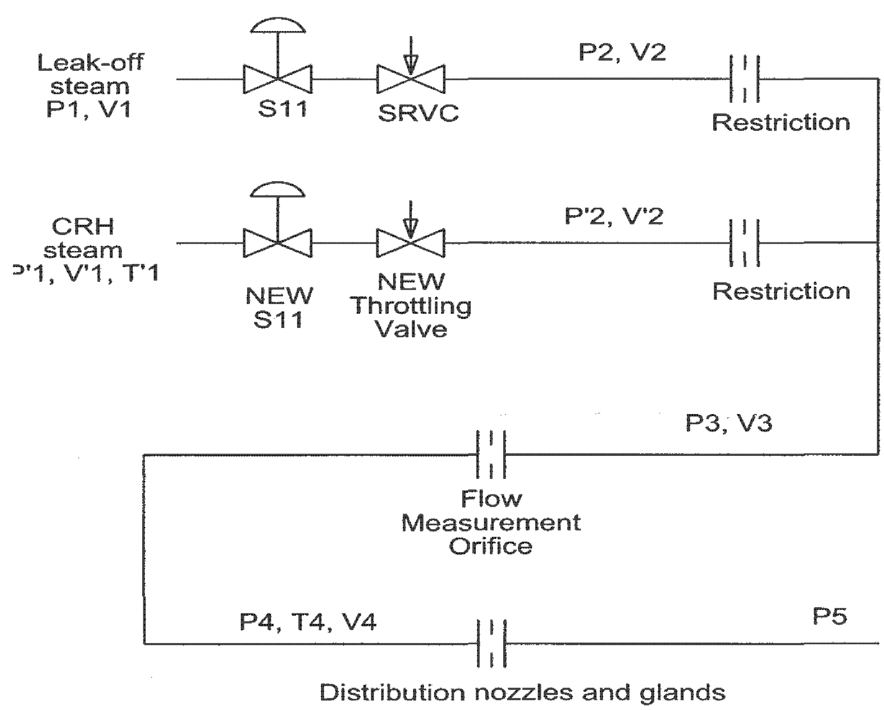
A method of reducing the net steam temperature is proposed, which takes a measure of steam from the HP cylinder exhaust in such a way that the mixed steam temperature can be controlled to pre-retrofit levels.

The proposal observes the requirements for IP rotor warming from the gland sealing system by means of the introduction of a pneumatically operated valve which prevents cool steam from the HP cylinder from reducing the overall warming effect. This valve will be operated by the same pneumatic signal as GE valve S11.

The effects of overspeed from the additional steam flow are taken into account by reducing the bore of the 0.9" restriction orifice in the existing cooling steam line.

2.0 Modelling the system.

The system is modelled as two parallel supply lines entering a common distribution system upstream of the flow measuring orifice. The capacity of the system should be such that we can achieve a flow of 11600 lbs/hour, plus a suitable margin for adjustment without exceeding the existing maximum flow following a load rejection. In order to do this, the system must be modelled as a combination of flow resistances using site data (taken 15 April 2003) to calculate the existing parameters. The convention is shown below:



Date: 23.8.04

2.1 IP Distribution nozzles and Glands

The low pressure drop downstream of the flow measuring orifice means that this part of the system can be modelled as:

$$Q = K4\sqrt{\frac{(P4 - P5)}{V4}}$$

where P5 is unknown.

V4 takes into account the warming of the steam as it flows over the central gland area, by assuming an average of the inlet temperature (downstream of the FMO) and the measured steam temperature at the first stage wheel.

In the first instance, (P4-P5)/V4 is plotted against the square of the flow and P5 varied until a least squares fit of the data results in a straight line passing through the origin. This results in a flow factor, K4, of 12432 and a mean P5 of 454.3 psi.

(Check: From GE Cooling Steam Diagram 148D5583, a first stage wheel pressure of 442.3 psi is shown to correspond with an inlet belt pressure of 531.2 psi.

At the time of the test, the pressure before the stop valve is recorded as 555.8 psi, therefore the estimate of P5 is consistent with an inlet belt pressure of (531.2*454.3/442.3) = 545.6 psi. This is equivalent to a pressure loss of approximately 10 psi over the chests and loop pipe which is within the expected value of 2% rated pressure. This result is acceptable.)

2.2 Flow Measuring Orifice

The FMO is modelled using the following equation:

$$Q = C.\frac{\pi}{4}.d^2\sqrt{\frac{(P3 - P4)}{V3}}.Y3 \quad \text{(metric data)}$$

where d is the bore of the orifice at operating conditions and the C factor is the coefficient of discharge of the orifice modified to take into account the velocity of approach. The expansion factor is derived from the explicit equation:

$$Y = 1 - (0.41 + 0.35.\beta^4).\frac{\Delta p}{p.\gamma}$$

2.3 Restriction Orifice Plates

The restriction orifice plate in the existing cooling line will always have a critical pressure drop across it. The new orifice plate (in the CRH supply) will normally exhibit a sub critical flow characteristic, but will have a critical pressure drop following load rejection (as the IP cylinder pressure decays). The following equations are used:

$$Q = 1.4103 \times 10^{-4} . d^2 \sqrt{\frac{P2}{V2}}$$ for critical flow (metric data)

or

The flow measuring orifice equation for sub-critical flow.

<u>2.4 Regulating Station</u> (existing installation)

In a similar manner to the IP nozzles and glands, the flow coefficient of the system comprising S11 and SRCV is modelled from the equation:

$$Q = K1\sqrt{\frac{(P1 - P2)}{V1}}.Y1$$

where Y1 is an expansion factor assumed to be of the order of 0.96. P2 is the pressure upstream of the restriction, and is calculated using the restriction orifice equation.

Since we are interested in the maximum capability of the system, the site data point with SRCV fully open is considered. This yields a P2 of 1296.7 psi and a K1 factor of 1117.1 .

<u>2.5 New Regulating Valve</u>

The new isolating valve will be a parallel slide valve with minimal pressure drop. The regulating valve will be a throttling valve and is assumed to have a Cg of 388 (which is consistent with 1" type 'A' valve in the Fisher catalogue). The Fisher gas sizing equation is used in establishing the pressure drop across this valve.

<u>3.0 Derivation of Results</u>

An iterative process has been used to establish orifice plate sizes that will yield acceptable mixing flows during operation without exceeding an acceptable flow following a load rejection. Because of the iterative nature of the calculation, the steam enthalpy and pressure is used to estimate the steam specific volume from the Callendar equation,

$V = 1.253 \times \frac{(h - 835)}{P}$. This gives results which are accurate to 1% over the range under consideration and will yield flow calculation results accurate to 0.5%.

3.1 Load Rejection

Following a load rejection, the HP and IP governing valves will rapidly close in. The steam within the HP casing will rapidly expand to CRH pressure and remain 'bottled' in the reheater circuit by virtue of the closed IP valves.

It is thus necessary to ensure that the combined flow from the modified system will not exceed the previous flow, which was limited by the 0.9" bore flow restrictor.

The existing limiting value can be calculated from the site data (regulating valve fully open) and the conditions that will exist following load rejection.

From site data, the maximum flow (SRCV fully open) is measured at the FMO and we are able to calculate the pressure upstream of the restriction orifice using the orifice critical pressure drop equation.

In a similar manner, following load rejection the upstream pressure (in the HP turbine casing) will fall to 624 psi and we can calculate the overspeed contributory flow using the K factor for the regulating station and the equation for critical flow through an orifice.

This yields a flow of 12456 lbs/hr, with 494 psi upstream of the restriction orifice.

Thus the modified system should not contribute to more than 12456 lbs/hr.

3.2 Normal Running.

In order to reduce the temperature of the cooling steam to pre-retrofit levels we require 23% of the cooling steam flow to be from HP exhaust. This is simply derived by finding the mean enthalpy of the mixed steam.

In addition, we require a mixed steam flow of at least 11,600 lbs/hr so that the pre-retrofit conditions can be restored.

By an iterative process the following orifice plate sizes are calculated:

2-1/2" line - Orifice bore to be 0.547"
1" line- - Orifice bore to be 0.645"

(Orifice bores calculated on the basis of expansion to 0.55" and 0.647" respectively.)

4.0 Calculation Results

The following details the various parameters that have been calculated in accordance with the foregoing model description.

4.1 During Normal Operation, with Regulating Valves fully open

Bled Steam Supply

P1 = 1664 psi, T1 = 944 °F, V1 = 0.465 ft^3/lb
P2 = 1575.5 ps1, V2 = 0.486 ft^3/lb
P3 = 477.5 psi
Flow = 13045 lb/hr

CRH Supply

P1′ = 595 psi, T1′ = 628 °F, V1′ = 1.001 ft^3/lb
P2′ = 559.8 psi, V2′ = 1.065 ft^3/lb
P3 = 477.5 psi
Flow = 4148 lb/hr

Flow Measuring Orifice

P3 = 477.5 psi, P4 = 457.5 psi, V3 = 1.518 ft^3/lb
Flow = 17193 lb/hr

4.2 Following Overspeed, with Regulating Valves fully open:

Bled Steam Supply

P1 = 624 psi, T1 = 628 °F, V1 = 0.952 ft^3/lb
P2 = 597.7 ps1, V2 = 0.993 ft^3/lb
Flow = 5630 lb/hr

CRH Supply

P′1 = 624 psi, T′1 = 628 °F, V′1 = 0.952 ft^3/lb
P′2 = 523.4 ps1, V′2 = 1.134 ft^3/lb
Flow = 6822 lb/hr

5.0 Conclusion

The proposed installation has a resized restriction orifice such that the maximum combined flow on load rejection does not exceed the current value of 12456 lb/hr.

During normal operation, the maximum combined flow equates to 17193 lb/hr which gives a margin of some 48% over the envisaged flow requirement.

Date: 23.8.04

Appendix:

Installation Requirements

The proposed arrangement for the modification is shown in figure A1 which shows all necessary piping falls for system drainage during start-up.

In a manner similar to the existing drain line, the new supply line should be supported at 2 points approximately equidistant along its length. The proposed support locations are shown on the diagram, however the final location should be selected to coincide with existing steelwork.

The final run of the line should be manufactured from low chrome molybdenum steel because of the possibility of heat conduction up to 850°F under low flow conditions. The flange pair is selected as an appropriate cut –off point.

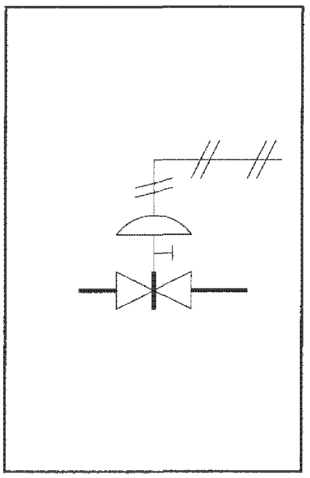
Envisaged insulation requirements are 2-1/2" thick preformed sections with appropriate aluminium cleading c/w two valve boxes and one flange box.

The bill of material is as follows:

Description	Quantity	Designation	Material
Sockolet at CRH:		36 – 12 x 1" – 3000lbs	ASTM A105
Pipe	30 ft	1" ns sched XS pipe	ASTM A106 grade B
Isolating Valve	1 off	see attached data sheet	
Regulating Valve	1off	800 lb, 1" globe needle valve. Flow coefficient to be greater than a Cv of 10.1	ASTM A105
Flanges	2 off	600 lb, 1" ns raised face slip on c/w bolting (excess length to accommodate orifice plate) and spiral wound joints	ASTM A182 grade F11
orifice plate	1 off	suitable for 1", 6-00 lb flange drilling - 0.645" bore	316 or 304 stainless
orifice plate	1 off	To suit existing 2-1/2" arrangement. – 0.547" bore	316 or 304 stainless
pipe	3 ft	1" ns sched XS pipe	ASTM A335 grade P11
Sockolet at IP rotor cooling line	1 off	5-4 x 1" – 3000lbs	ASTM A182 grade F11

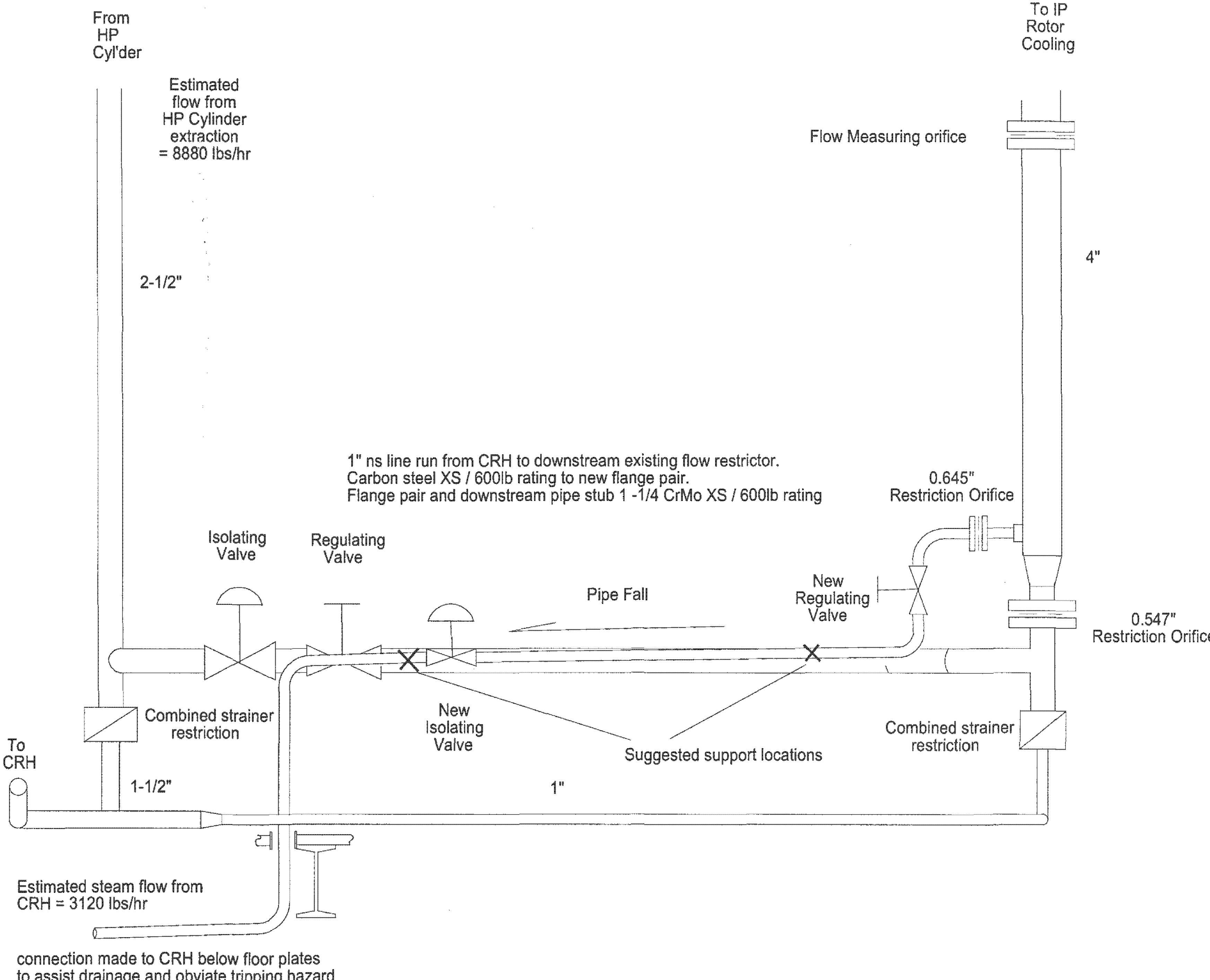
Date: 23.8.04

Pneumatic Valve Data Sheet (Parallel Slide Type)

Iss			
	No. per TG Set 1		**Contract** SIDI KRIR
	Supplier #		Valve Number TBA
	Valve Body Data		Label:-
	Body Size/Type No. 1"/#		
	Trim Size inch #		
	Body Material/Spec. C.S. ASTM A105		
	Body Form/Rating Butt weld/class 600lb		
	Style Parallel Slide Gate		
	Non-rotating, rising stem with opening backseating		
	Trim Material # 13Cr + Stellite		
	Packing # Graphite		
	Body Des. Temp./Press. 630°C/630 psi		
	Inlet/Outlet Connection Size # BW-XS		
	Fluid Data/Performance		
	Line Fluid Steam		
	Design Flow 4000 lb/hr		
	Press drop at design flow #		
	Press. Drop Shutoff for Actuator Sizing 630 psi		
	Calc. CV/Valve CV #/#		
	Noise level at design flow # dBA (Max 85 dBA)		
	Line Data		**Supplementary Design Conditions**
	Line Size (See Note) 1" NB		Air Supply Pressure 100 psi
	Upstream 1.314" OD x 0.179" wall		Instrument Air Consumption
	Downstream 1.314" OD x 0.179" wall		Steady State: # Sm^3/hr Transient: # Sm^3/hr
	Actuator Data		
	Actuator Type/Size Pneumatic #		**Accessories Data**
	Actuator Form Diaphragm		
	Spring Range #		
	Spring Tends To Open Valve		
	Air Signal Close Valve		
	Supply Air Failure Open		
	Handwheel Yes		
	I/P None		
	I/P Action N/A		
	Positioner None		
	Valve Action N/A		
	Positioner Action N/A		
	Filter/Regulator No/Type No #		
	Airlock No		
	Booster No		
	Stroke Time Open/Closed #		**Installation**
			Height mm # Width mm #
			Length mm # Mass kg #

NOTES:- # Information supplied by supplier. SCOPE OF SUPPLY:- Supply as complete Assy. with Single Termination Point for Air Pipework (1/4" NPTF).

Date: 23.8.04


From HP Cyl'der
Estimated flow from HP Cylinder extraction = 8880 lbs/hr
2-1/2"
To IP Rotor Cooling
Flow Measuring orifice
4"
1" ns line run from CRH to downstream existing flow restrictor.
Carbon steel XS / 600lb rating to new flange pair.
Flange pair and downstream pipe stub 1 -1/4 CrMo XS / 600lb rating
0.645" Restriction Orifice
Isolating Valve
Regulating Valve
Pipe Fall
New Regulating Valve
0.547" Restriction Orifice
Combined strainer restriction
New Isolating Valve
Suggested support locations
Combined strainer restriction
To CRH
1-1/2"
1"
Estimated steam flow from CRH = 3120 lbs/hr
connection made to CRH below floor plates to assist drainage and obviate tripping hazard

IP7008466

From: <robert.cunningham@power.alstom.com>
To: <phong-d@ipsc.com>
Date: 9/9/2003 7:00:45 AM
Subject: Intermountain Creep Life Assessment

Hello Phong

Please find below our Creep Life Assessment of the IP Rotor.

The effect of having a 947oF cooling steam flow has been assessed in terms of the rate of creep life usage for the IP rotor. The critical locations are expected to be the blade root fastening in the disc head at stage 1 and the rotor bore beneath the stage 1 disc.

We have made the following assumption for reference points:-
- At the Alarm Point setting of 937oF, the rotor bore and stage 1 root have a creep life usage factor of 1.0.

The reference point of the stage 1 root is based on GE Instruction GEK-72307A 'Reheat cooling steam system'. This instruction describes that the steam flow through the passages in the stage 1 blade roots will consist of a mixture of the cooling steam flow and steam entrained from the main steam flow, with the passage flow being 50% cooling steam and 50% entrained main steam.

The assessment of the stage 1 root has taken into account the increased flow from the IP rotor cooling steam with the passage flow being 100% cooling steam. From the reference point to the new cooling steam flow condition we predict an increase of 14oF at the root.

The assessment of the rotor bore has been based on the assumption that the operating temperature has increased by the same amount as the increase in the measured effective cooling steam temperature i.e. 10oF.

The effect of the increase in rotor metal temperature has been assessed by reference to the results of extensive long term multi-temperature creep rupture testing of samples from 1%CrMoV rotor forgings.

> For the stage 1 root, these test indicate that a temperature increase of 14oF corresponds to a shortening of rupture life by a factor of 0.67. This indicates that the rate of creep life usage in the IP rotor is increased by a factor of 1.49 when operating with the maximum measured effective cooling steam temperature of 947 oF, i.e. 1,000 hours operation in this condition corresponds to 1,490 hours operation at design conditions. The impact of this on the overall creep life of the rotor is dependent on the design life. Assuming a design life of 250,000 hours the penalty of operating for a limited period, say 13,000 hours (18 months) in the current condition would be about 2.5% (i.e. an extra 2.5% of the design life of the rotor would be consumed during this period).
> The effect on the creep life of the rotor bore is expected to be less than this, as the temperature increase is less.

I hope this information will be of use to you, for making the your proposal.

Regards

Rob C

:.________________

CC: <kevin.spires@power.alstom.com>, <tim.shurrock@power.alstom.com>, <simon.glover@power.alstom.com>, <RUGWW.TEGMail@test.alstom.com>

From: <robert.cunningham@power.alstom.com>
To: <phong-d@ipsc.com>
Date: 4/30/03 10:28AM
Subject: Intermountain IP Rotor Cooling Temperature

Hello Phong

Have you had any luck finding the P, T and flows before the Retrofit Outage??

Alan will be sending to you today the specs, ratings etc of the pipework, flanges, valves, etc that are required for the mod.

We will inclue a regulating valve in the new line.

Question about 1 of your photos. It woul appear that the take off from the CRH goes down, rather than up, after coming of the CRH. Can you please confirm that this is the case.

Regards

Rob C

:.___________________

CC: <alan.lang@power.alstom.com>, <alain.foote@power.alstom.com>, <RUGWW.TEGMail@test.alstom.com>

From: <robert.cunningham@power.alstom.com>
To: <phong-d@ipsc.com>
Date: 5/8/03 9:18AM
Subject: Intermountain IP Rotor Cooling

Hello Phong,

Please find below:-

a sketch showing a layout suggestion for the CRH top-up connection.
a proposed set-up procedure for the CRH top-up connection.

Layout Suggestion :- (See attached file: new layout.tif)

The predicted flows to obtain the original level as cooling from the HP cylinder (after stage 2) and the from the CRH are identified on the sketch. These values are not limited by the system and will be controlled by the regulating vavles.

Note : The connection withn the CRH is deliberately below floor level so that we don't have a tripping hazard whilst maintaining adequate drainage.

Proposed Set-up Procedure

Following installation of a CRH top-up system to allow regulation of IP rotor cooling steam outlet temperature the following set-up procedure is recommended:

1. Prior to start up:-
   1.1 The existing rotor cooling manual regulating valve (SRCV) should be fully open.
   1.2 The newly installed regulating valve should be cracked open (approx 1/8 turn) to allow the line to be warmed through at start up.
   1.3 Suitable instrumentation should be fitted to permit measurement of flow from the 2" flow measuring orifice.

2. Once full (or near full) power is reached, SRCV should be gradually closed in to give a differential pressure of 10.3 psi at the orifice plate, corresponding to a flow of approximately 12000 lbs/hr.

3. The new regulating valve should be gradually opened to give the correct temperature at the eighth stage wheel. (This will have the effect of increasing th enet flow by about 30%).

4. Gradually close in at SRCV to give 12.3 psi DP at the flow measuring orifice. (At this point the CRH steam concentration will be slightly high, so subsequent reduction of flow is required).

5. Gradually close in the new regulating valve to give required eighth stage wheel steam temperature measurements.

I hope this information will be of use to you.

Regards

Rob Cunningham

:.________________

CC: <alain.foote@power.alstom.com>, <alan.lang@power.alstom.com>, <robert.brown@power.alstom.com>, <alan.holmes@power.alstom.com>, <tim.shurrock@power.alstom.com>, <RUGWW.TEGMail@test.alstom.com>

IPSC
INTERMOUNTAIN POWER SERVICE CORPORATION

QUALITY ASSURANCE CHECK LIST FOR IP TURBINE 8TH STAGE DIAPHRAGM REBUILD

General

The IGS IP turbine 8th stage diaphragm is a double flow with the TE removable from the GE. The diaphragm has FOD, SPE and spill strip damage and requires a major rebuild. It is currently at the MDA Turbine Blading Repair shop for rebuilding and coating.

QA and visual inspection on the partition profile, throat area, horizontal joint gap, seal face gap and surface roughness are important to assure proper reliability and performance. Inspection includes:

a. Erosion, deposits and mechanical damage on diaphragms, diaphragms packings, tip radial spill strips and ends packings.
b. Control performance losses for each steam component to minimize leakages, minimize friction, aerodynamic changes and control state energy distribution.
c. Ensure horizontal joint and TE/GE seal face check. Improper seal face would cause higher IP cooling steam cooling.

QUALITY ASSURANCE CHECK LIST
FOR IP TURBINE 8TH STAGE DIAPHRAGM REBUILD

Process Control Sheets

No	Activity Description	QA Instructions References	Verifier Date
1	As found inspection		
2	As found and as left measurements of diaphragm: roundness, flatness, dishing, hj, seal faces to seal face and crush pin height, gap check, diameter check, drop check		
3	As found measurements of throat and partition		
4	Final measurement of throat opening including radial heights, throat readings, pitch readings and individual areas		
5	Final trailing edge thickness measurement		
6	Final partition to sidewall joint measurement		
7	Final contour measurement		
8	Calculated/measured of the individual throat areas		
9	Calculated/measured of the total diaphragm throat areas		
10	Design vs as found total diaphragm throat areas		
11	Weld procedures and documentation		
12	Stress Relief documentation		
13	Pre-Repair NDE report and review		
14	Post-Repair NDE report and review		
15	Packing rings and spill strips tooth clearance		
16	Packing rings and spill strips butt clearance		

QUALITY ASSURANCE CHECK LIST
FOR IP TURBINE 8TH STAGE DIAPHRAGM REBUILD

No	Activity Description	QA Instructions References	Verifier Date
17	As left inspection check list		

As Found Inspection

DIAPHRAGM INSPECTION CHECKLIST	STAGE 8 TE UPPER	STAGE 8 GE UPPER	STAGE 8 TE LOWER	STAGE 8 GE LOWER
As found measurements of diaphragm: roundness, flatness, dishing, hj, seal face to seal face and crush pin height				
Blast Clean				
NDE Entire Diaphragm				
Inspect for FOD				
Inspect for Erosion				
Inspect Joints Dowels				
Inspect Spill Strips				
Inspect Support Bars				
Inspect Packing Areas				
Inspect Horizontal Joint				
All Threaded Holes				
Inspect Seal Face				
Repair Completed				
Inspect Crush Pins				
Snout Ring Bore Hone and Clean				
Surface roughness				
Bonding Strength of the Final Coating				

QUALITY ASSURANCE CHECK LIST
FOR IP TURBINE 8TH STAGE DIAPHRAGM REBUILD

DIAPHRAGM INSPECTION CHECKLIST	STAGE 8 TE UPPER	STAGE 8 GE UPPER	STAGE 8 TE LOWER	STAGE 8 GE LOWER
V- Visual Inspection PT- Penetrans NA - Not Applicable	MP - Mag. Particle X - Work Carried Out C - See Comments		UT - Ultrasonic N - Not Done	
Completed By:		Date:		

As Left Inspection

DIAPHRAGM INSPECTION CHECKLIST	STAGE 8 TE UPPER	STAGE 8 GE UPPER	STAGE 8 TE LOWER	STAGE 8 GE LOWER
As left measurements of diaphragm: roundness, flatness, dishing, hj, seal face to seal face and crush pin height				
Blast Clean				
NDE Entire Diaphragm				
Inspect for FOD				
Inspect for Erosion				
Inspect Joints Dowels				
Inspect Spill Strips				
Inspect Support Bars				
Inspect Packing Areas				
Inspect Horizontal Joint				
All Threaded Holes				
Inspect Seal Face				
Repair Completed				
Inspect Crush Pins				
Snout Ring Bore Hone and Clean				

QUALITY ASSURANCE CHECK LIST
FOR IP TURBINE 8TH STAGE DIAPHRAGM REBUILD

DIAPHRAGM INSPECTION CHECKLIST	STAGE 8 TE UPPER	STAGE 8 GE UPPER	STAGE 8 TE LOWER	STAGE 8 GE LOWER
Final Surface Roughness				
Bonding Strength of the Final Coating				
Bolt Up the 2 half at full torque value then check horizontal join gap. Should be less than 2 mils				
Bolt Up the 2 half at full torque value and when the hori'l join gap is within spec, blue check the steam seal face of the upper and lower, both GE and TE.				
V- Visual Inspection PT- Penetrant NA - Not Applicable	MP - Mag. Particle X - Work Carried Out C - See Comments	UT - Ultrasonic N - Not Done		
Completed By:		Date:		

From: <alan.lang@power.alstom.com>
To: <robert.cunningham@power.alstom.com>
Date: 4/30/2003 11:18:58 AM
Subject: Re: Intermountain IP Rotor Cooling Temperature

Rob,

The mixture of steam will be in approximately the ratio 26% CRH steam and 74% stage extraction steam.

A 1" line will be adequate to furnish this whether we go for the flows at present or a more realistic 11600 lbs/hr.

The original intention was to supply an orifice plate which would take the bulk of the pressure drop however a regulating valve will cover this duty without undue problem.

I recommend that we use a new 1" tapping into the CRH so that there are no concerns about water entrainment. This can run alongside the existing drain, rising to the elevation of the existing regulating valve to allow side by side location. The check valve should, however, be located at the highest point (just upstream of the connection with the existing piping above the existing restriction orifice) so that there is no danger of accumulating a slug of condensate above it.

All piping should be 1" schedule XS. This will be carbon steel up to the check valve with carbon steel class 600lb (or 800 lb socket weld) regulating globe valve.

The check valve, (either screw down non return, lift check or swing check style) should also be 600lb rating in alloy steel (F22, 2-1/4 chrome - moly) and the piping/fittings downstream of this should also be grade P22/F22 as appropriate. This 600 lb rating is based on the max available pressure ~540 psi at 940 deg F.

Suggest we use sockolet connections into the existing piping.

Regards,

Alan

:.____________________

CC: <phong-d@ipsc.com>, <alain.foote@power.alstom.com>, <RUGWW.TEGMail@test.alstom.com>

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 9/1/2004 7:55:58 AM
Subject: Re: IP Clearance Data

Hi Phong,

I see nothing wrong with your clearance data. We also believe that the problem may be due to steam leakage past the first stage diaphragm on the steam end flow.

Attached is my preliminary calculation report on the orifice sizing and system installation. Would you please review to see if this satisfies your immediate needs.

One thing I am not clear about is your standard mode of operation - your guidance would be useful.

I assume that for cold starts HSPV is used to pressurise the HP casing and warm it through by condensation heating. This would normally take place whilst condenser vacuum is being raised, at which point the IP rotor will also be heated through HSFV from the gland packing lines.

Once the turbine is ready to roll, HSPV will close and the HP cylinder will depressurise.

HSFV will then close and S11 will open.

If I am correct on the above assumption then I foresee no problem. If, however, S11 is programmed to open before the HP cylinder depressurises then there may be a drainage problem and we may need to consider a drain orifice to bypass the new pneumatic isolating valve.

Note also, do you believe it would be necessary to supply this valve with limit switches? These could be arranged in parallel to the limit switches on S11 to signify correct operation.

An estimate of the performance benefit and the mark-up to 903E892 will follow.

Regards,

Alan

(See attached file: IPS0001A2.pdf)

CC: <kevin.spires@power.alstom.com>,
<robert.cunningham@power.alstom.com>

Intermountain

IP Rotor Cooling Augmentation Calculation

CONTENTS

1. Introduction

2. Modelling The System

3. Derivation of Results

4. Calculation Results

5. Conclusion

Appendix

Date: 23.8.04

Issue: A

1.0 Introduction

Following retrofit of the HP cylinder on Intermountain unit 1, excessively high temperatures have been observed at the first stage wheel of the IP cylinder. This is because of high enthalpy steam from the HP second stage tip seals being introduced into the leak-off cooling flow to the IP rotor cooling system.

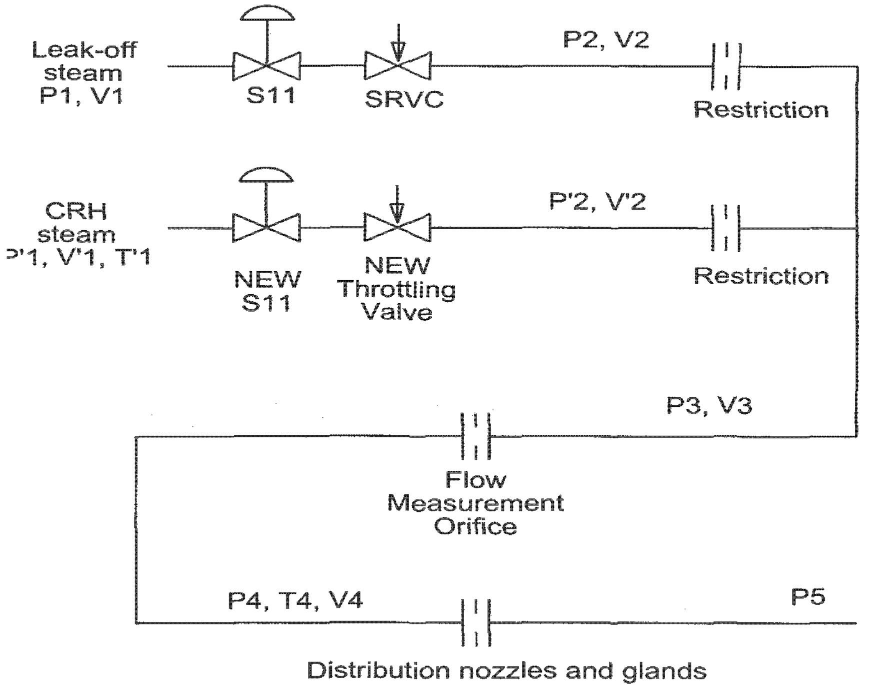
A method of reducing the net steam temperature is proposed, which takes a measure of steam from the HP cylinder exhaust in such a way that the mixed steam temperature can be controlled to pre-retrofit levels.

The proposal observes the requirements for IP rotor warming from the gland sealing system by means of the introduction of a pneumatically operated valve which prevents cool steam from the HP cylinder from reducing the overall warming effect. This valve will be operated by the same pneumatic signal as GE valve S11.

The effects of overspeed from the additional steam flow are taken into account by reducing the bore of the 0.9" restriction orifice in the existing cooling steam line.

2.0 Modelling the system.

The system is modelled as two parallel supply lines entering a common distribution system upstream of the flow measuring orifice. The capacity of the system should be such that we can achieve a flow of 11600 lbs/hour, plus a suitable margin for adjustment without exceeding the existing maximum flow following a load rejection. In order to do this, the system must be modelled as a combination of flow resistances using site data (taken 15 April 2003) to calculate the existing parameters. The convention is shown below:



2.1 IP Distribution nozzles and Glands

The low pressure drop downstream of the flow measuring orifice means that this part of the system can be modelled as:

$$Q = K4\sqrt{\frac{(P4 - P5)}{V4}}$$

where P5 is unknown.

V4 takes into account the warming of the steam as it flows over the central gland area, by assuming an average of the inlet temperature (downstream of the FMO) and the measured steam temperature at the first stage wheel.

In the first instance, (P4-P5)/V4 is plotted against the square of the flow and P5 varied until a least squares fit of the data results in a straight line passing through the origin. This results in a flow factor, K4, of 12432 and a mean P5 of 454.3 psi.

(Check: From GE Cooling Steam Diagram 148D5583, a first stage wheel pressure of 442.3 psi is shown to correspond with an inlet belt pressure of 531.2 psi.

At the time of the test, the pressure before the stop valve is recorded as 555.8 psi, therefore the estimate of P5 is consistent with an inlet belt pressure of (531.2*454.3/442.3) = 545.6 psi. This is equivalent to a pressure loss of approximately 10 psi over the chests and loop pipe which is within the expected value of 2% rated pressure. This result is acceptable.)

2.2 Flow Measuring Orifice

The FMO is modelled using the following equation:

$$Q = C.\frac{\pi}{4}.d^2\sqrt{\frac{(P3 - P4)}{V3}}.Y3 \quad \text{(metric data)}$$

where d is the bore of the orifice at operating conditions and the C factor is the coefficient of discharge of the orifice modified to take into account the velocity of approach. The expansion factor is derived from the explicit equation:

$$Y = 1 - (0.41 + 0.35.\beta^4).\frac{\Delta p}{p.\gamma}$$

2.3 Restriction Orifice Plates

The restriction orifice plate in the existing cooling line will always have a critical pressure drop across it. The new orifice plate (in the CRH supply) will normally exhibit a sub critical flow characteristic, but will have a critical pressure drop following load rejection (as the IP cylinder pressure decays). The following equations are used:

$$Q = 1.4103 \times 10^{-4} . d^2 \sqrt{\frac{P2}{V2}}$$ for critical flow (metric data)

or

The flow measuring orifice equation for sub-critical flow.

2.4 Regulating Station (existing installation)

In a similar manner to the IP nozzles and glands, the flow coefficient of the system comprising S11 and SRCV is modelled from the equation:

$$Q = K1\sqrt{\frac{(P1-P2)}{V1}}.Y1$$

where Y1 is an expansion factor assumed to be of the order of 0.96. P2 is the pressure upstream of the restriction, and is calculated using the restriction orifice equation.

Since we are interested in the maximum capability of the system, the site data point with SRCV fully open is considered. This yields a P2 of 1296.7 psi and a K1 factor of 1117.1 .

2.5 New Regulating Valve

The new isolating valve will be a parallel slide valve with minimal pressure drop. The regulating valve will be a throttling valve and is assumed to have a Cg of 388 (which is consistent with 1" type 'A' valve in the Fisher catalogue). The Fisher gas sizing equation is used in establishing the pressure drop across this valve.

3.0 Derivation of Results

An iterative process has been used to establish orifice plate sizes that will yield acceptable mixing flows during operation without exceeding an acceptable flow following a load rejection. Because of the iterative nature of the calculation, the steam enthalpy and pressure is used to estimate the steam specific volume from the Callendar equation,

$V = 1.253 \times \frac{(h-835)}{P}$. This gives results which are accurate to 1% over the range under consideration and will yield flow calculation results accurate to 0.5%.

3.1 Load Rejection

Following a load rejection, the HP and IP governing valves will rapidly close in. The steam within the HP casing will rapidly expand to CRH pressure and remain 'bottled' in the reheater circuit by virtue of the closed IP valves.

It is thus necessary to ensure that the combined flow from the modified system will not exceed the previous flow, which was limited by the 0.9" bore flow restrictor.

The existing limiting value can be calculated from the site data (regulating valve fully open) and the conditions that will exist following load rejection.

From site data, the maximum flow (SRCV fully open) is measured at the FMO and we are able to calculate the pressure upstream of the restriction orifice using the orifice critical pressure drop equation.

In a similar manner, following load rejection the upstream pressure (in the HP turbine casing) will fall to 624 psi and we can calculate the overspeed contributory flow using the K factor for the regulating station and the equation for critical flow through an orifice.

This yields a flow of 12456 lbs/hr, with 494 psi upstream of the restriction orifice.

Thus the modified system should not contribute to more than 12456 lbs/hr.

3.2 Normal Running.

In order to reduce the temperature of the cooling steam to pre-retrofit levels we require 23% of the cooling steam flow to be from HP exhaust. This is simply derived by finding the mean enthalpy of the mixed steam.

In addition, we require a mixed steam flow of at least 11,600 lbs/hr so that the pre-retrofit conditions can be restored.

By an iterative process the following orifice plate sizes are calculated:

2-1/2" line - Orifice bore to be 0.547"
1" line - - Orifice bore to be 0.645"

(Orifice bores calculated on the basis of expansion to 0.55" and 0.647" respectively.)

4.0 Calculation Results

The following details the various parameters that have been calculated in accordance with the foregoing model description.

4.1 During Normal Operation, with Regulating Valves fully open

Bled Steam Supply

P1 = 1664 psi, T1 = 944 °F, V1 = 0.465 ft^3/lb
P2 = 1575.5 ps1, V2 = 0.486 ft^3/lb
P3 = 477.5 psi
Flow = 13045 lb/hr

CRH Supply

P1' = 595 psi, T1' = 628 °F, V1' = 1.001 ft^3/lb
P2' = 559.8 psi, V2' = 1.065 ft^3/lb
P3 = 477.5 psi
Flow = 4148 lb/hr

Flow Measuring Orifice

P3 = 477.5 psi, P4 = 457.5 psi, V3 = 1.518 ft^3/lb
Flow = 17193 lb/hr

4.2 Following Overspeed, with Regulating Valves fully open:

Bled Steam Supply

P1 = 624 psi, T1 = 628 °F, V1 = 0.952 ft^3/lb
P2 = 597.7 ps1, V2 = 0.993 ft^3/lb
Flow = 5630 lb/hr

CRH Supply

P'1 = 624 psi, T'1 = 628 °F, V'1 = 0.952 ft^3/lb
P'2 = 523.4 ps1, V'2 = 1.134 ft^3/lb
Flow = 6822 lb/hr

5.0 Conclusion

The proposed installation has a resized restriction orifice such that the maximum combined flow on load rejection does not exceed the current value of 12456 lb/hr.

During normal operation, the maximum combined flow equates to 17193 lb/hr which gives a margin of some 48% over the envisaged flow requirement.

Date: 23.8.04

Issue: A

Appendix:

Installation Requirements

The proposed arrangement for the modification is shown in figure A1 which shows all necessary piping falls for system drainage during start-up.

In a manner similar to the existing drain line, the new supply line should be supported at 2 points approximately equidistant along its length. The proposed support locations are shown on the diagram, however the final location should be selected to coincide with existing steelwork.

The final run of the line should be manufactured from low chrome molybdenum steel because of the possibility of heat conduction up to 850°F under low flow conditions. The flange pair is selected as an appropriate cut –off point.

Envisaged insulation requirements are 2-1/2" thick preformed sections with appropriate aluminium cleading c/w two valve boxes and one flange box.

The bill of material is as follows:

Description	Quantity	Designation	Material
Sockolet at CRH:		36 – 12 x 1" – 3000lbs	ASTM A105
Pipe	30 ft	1" ns sched XS pipe	ASTM A106 grade B
Isolating Valve	1 off	see attached data sheet	
Regulating Valve	1off	800 lb, 1" globe needle valve. Flow coefficient to be greater than a Cv of 10.1	ASTM A105
Flanges	2 off	600 lb, 1" ns raised face slip on c/w bolting (excess length to accommodate orifice plate) and spiral wound joints	ASTM A182 grade F11
orifice plate	1 off	suitable for 1", 6-00 lb flange drilling - 0.645" bore	316 or 304 stainless
orifice plate	1 off	To suit existing 2-1/2" arrangement. – 0.547" bore	316 or 304 stainless
pipe	3 ft	1" ns sched XS pipe	ASTM A335 grade P11
Sockolet at IP rotor cooling line	1 off	5-4 x 1" – 3000lbs	ASTM A182 grade F11

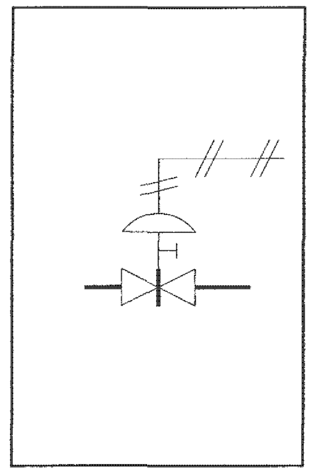
Date: 23.8.04

Issue: A

Pneumatic Valve Data Sheet (Parallel Slide Type)

Iss			
	No. per TG Set 1		
	Supplier	#	
	Valve Body Data		
	Body Size/Type No.	1"/#	
	Trim Size	inch #	
	Body Material/Spec.	C.S. ASTM A105	
	Body Form/Rating	Butt weld/class 600lb	
	Style	Parallel Slide Gate	
	Non-rotating, rising stem with opening backseating		
	Trim Material	# 13Cr + Stellite	
	Packing	# Graphite	
	Body Des. Temp./Press. 630°C/630 psi		
	Inlet/Outlet Connection Size	# BW-XS	
	Fluid Data/Performance		
	Line Fluid	Steam	
	Design Flow	4000 lb/hr	
	Press drop at design flow #		
	Press. Drop Shutoff for Actuator Sizing	630 psi	
	Calc. CV/Valve CV	#/#	
	Noise level at design flow # dBA (Max 85 dBA)		
	Line Data		
	Line Size (See Note)	1" NB	
	Upstream	1.314" OD x 0.179" wall	
	Downstream	1.314" OD x 0.179" wall	
	Actuator Data		
	Actuator Type/Size Pneumatic #		
	Actuator Form	Diaphragm	
	Spring Range	#	
	Spring Tends To	Open Valve	
	Air Signal	Close Valve	
	Supply Air Failure	Open	
	Handwheel	Yes	
	I/P	None	
	I/P Action	N/A	
	Positioner	None	
	Valve Action	N/A	
	Positioner Action	N/A	
	Filter/Regulator	No/Type No #	
	Airlock	No	
	Booster	No	
	Stroke Time Open/Closed	#	

Contract	SIDI KRIR		
Valve Number	TBA		
Label:-			



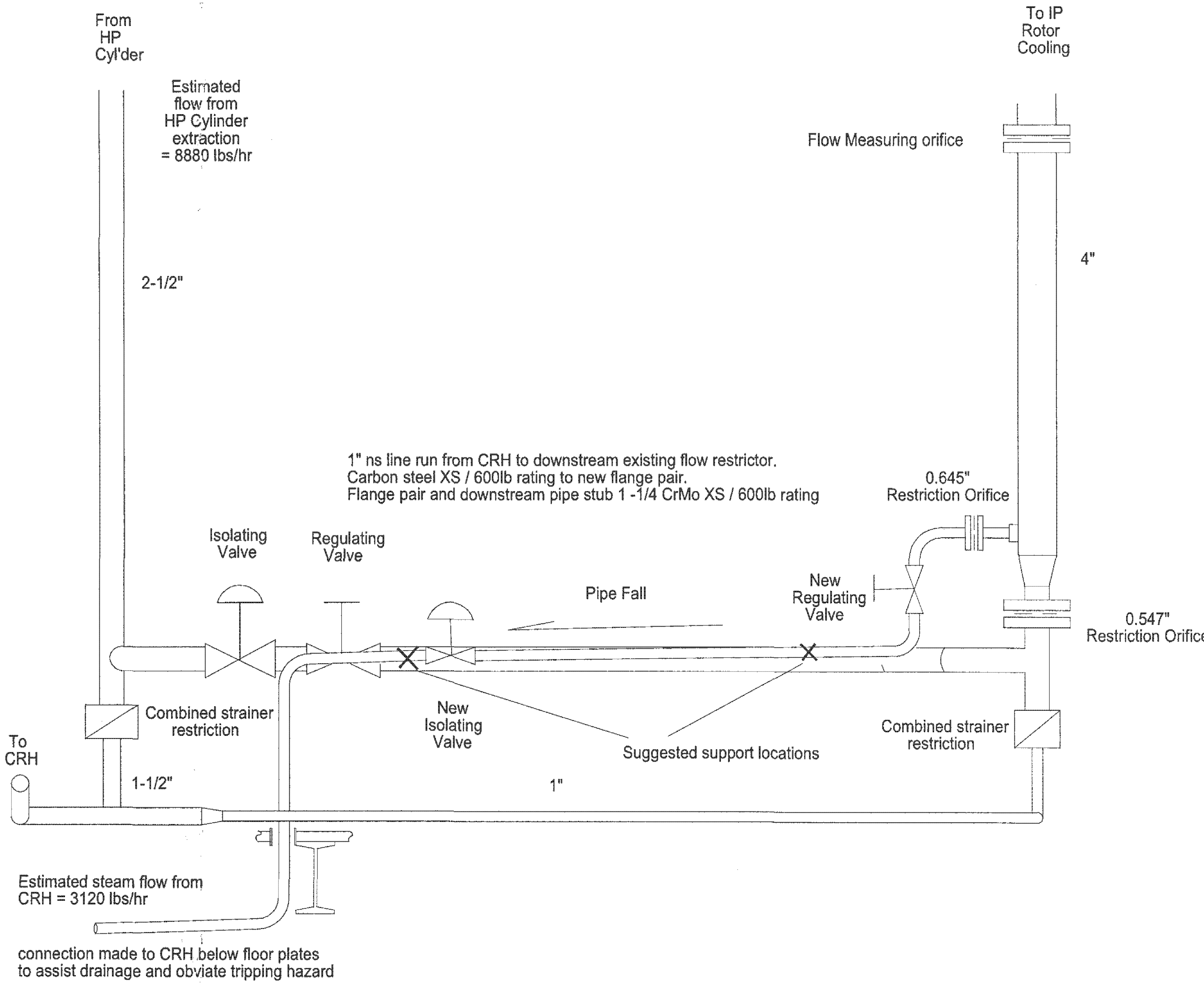
Supplementary Design Conditions			
Air Supply Pressure	100 psi		
Instrument Air Consumption			
Steady State: # Sm^3/hr	Transient: # Sm^3/hr		
Accessories Data			
Installation			
Height mm	#	Width mm	#
Length mm	#	Mass kg	#

NOTES:- # Information supplied by supplier. SCOPE OF SUPPLY:- Supply as complete Assy. with Single Termination Point for Air Pipework (1/4" NPTF).

Date: 23.8.04

Issue: A

IP7008488


From HP Cyl'der
Estimated flow from HP Cylinder extraction = 8880 lbs/hr
2-1/2"
To IP Rotor Cooling
Flow Measuring orifice
4"
1" ns line run from CRH to downstream existing flow restrictor.
Carbon steel XS / 600lb rating to new flange pair.
Flange pair and downstream pipe stub 1 -1/4 CrMo XS / 600lb rating
0.645" Restriction Orifice
Isolating Valve
Regulating Valve
Pipe Fall
New Regulating Valve
0.547" Restriction Orifice
Combined strainer restriction
New Isolating Valve
Suggested support locations
Combined strainer restriction
To CRH
1-1/2"
1"
Estimated steam flow from CRH = 3120 lbs/hr
connection made to CRH below floor plates to assist drainage and obviate tripping hazard

IP70008489

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 8/6/2004 6:43:02 AM
Subject: Re: IP Cooling Steam Reduction

Hi Phong,

I would like clarification for a few points in your latest e-mail as follows:

1/ We are unable to open attachment file 62.1001.05-10200-R002.CIT. Is this an important file? Could you please send a scanned image?

2/ Overspeed.

Without access to the OEM design calculation data, an overspeed calculation will be very difficult to carry out. The proposed additional cooling steam flow from CRH will be very low in comparison to flow through the IP stop valves during closure and other stored steam flow rates and I estimate that the overspeed contribution from this cooling flow will be 0.01% of rated speed. This is a relatively trivial amount and I hope that you will be prepared to accept a more rudimentary calculation which demonstrates this.

In order to assist on this front, an estimate of the peak turbine overspeed following load rejection and the time to reach this peak speed would be most appreciated. Otherwise we will need to use standard data based on our own experience.

3/ Interlocks.

It is envisaged that the additional cooling steam supply line will not be automatically operated since there is no operational mode in which the IP cylinder is run independently of the HP cylinder. Therefore it is intended that no interlocks be fitted - would you please clarify this point if you feel that more needs to be done. Note that the proposed check valve is intended to prevent leakage of steam during the warming phase (when warming valve HSFV is open). Is it possible that you would prefer to see a pneumatic valve which operates in a similar manner to rotor cooling valve S11 in place of the check valve ?

Best regards,

Alan

:.____________________

CC: <kevin.spires@power.alstom.com>,
<rob.cunningham@power.alstom.com>

From: Jim Knapp
To: Phong Do
Date: 8/18/2004 10:52:14 AM
Subject: Re: New S11 Valve for the New Line

Hello Phong,

TGC-ACV-19 is a single acting actuator. It recieves an air signal to go closed and uses spring action to open. The existing valve is a Fisher valve that is designed to be a pressure control valve. The signal that tells the valve to close comes from SV-40, which is an ASCO solenoid valve. The ASCO P/N is HT8320A185.

Without knowing more about what your plan is, it is difficult to know what to recommend. I'll be happy to help in any way I can.

Jim Knapp

>>> Phong Do 8/18/2004 7:10:26 AM >>>
Jim,
In an effort of reducing the U1 IP cooling steam temperature, we are considering:
1. Install new line which runs from HP cold reheat to the existing IP cooling line to mix and cool the steam.
2. Install new S11 and SRCV (ACV-19 and ACV-20, 1TGC-M4080A) on the new line for performance and overspeed protection.

Please let me know if the existing S11 (ACV-19) has a single acting or double acting actuator as this will affect the pneumatic interface with the new valve. It would be good if you can send me the spec of this actuator and solenoids.

I will go over with you when the design package is completed. Thank you much.

>>> <alan.lang@power.alstom.com> 8/18/2004 1:30:59 AM >>>
Hello Phong,

We agree to the need for an isolating valve in the new cooling line since steam from the pressurised HP cylinder will be less effective at warming the IP rotor than steam from the gland sealing line.

We propose to use a parallel slide gate valve as this will require less operating force than a globe valve - is this OK?

Would you also find out if the existing S11 has a single acting or double acting actuator as this will affect the pneumatic interface with the new valve.

Thank you and best regards,

Alan

:.________________

Mime-Version: 1.0
X-Mailer: Groupwise 6.5
Subject: RE: Response Letter to ABT
Date: Thu, 10 Aug 2006 06:54:20 -0600
Message-ID: <44DAD81C0200000B0011AEDF@ipsc.com>
References: <44D8FF79020000060015C358@ipsc.com>
In-Reply-To: <44D8FF79020000060015C358@ipsc.com>
From: "Jerry Hintze" <JERRY-H@ipsc.com>
To: "Jason W. Hardin" <jhardin@fabianlaw.com>
Content-Type: text/plain; charset=us-ascii
Content-Transfer-Encoding: quoted-printable
Content-Disposition: inline

Jason, Don't bother spending a lot of time on the letter because somehow =
it got already sent to them even though I was holding the original.   I =
will send you their response assuming they will respond. =20

>>> "Jason W. Hardin" <jhardin@fabianlaw.com> 08/08 9:14 PM >>>
Jerry,

I received your voice mail earlier today. I will review and analyze the draft letter and send you my comments as soon as I can, no later than the end of the week. I will try to get to it tomorrow, but, frankly, because I am under the gun on a couple of other cases, I probably will not get to it until Thursday. Please call or email me if that time frame is unacceptable.

Thanks,
Jason

-----Original Message-----
From: Jerry Hintze [mailto:JERRY-H@ipsc.com]=20
Sent: Monday, July 31, 2006 10:04 AM
To: Jason W. Hardin
Cc: Dean Wood; Dennis Killian; Garry Christensen; George Cross
Subject: Response Letter to ABT

Jason,

Attached is a draft or our proposed response to ABT's letter of May 9, 2006. Please review and give me your comments. If you have any questions, you can contact either myself, Dean Wood or Garry Christensen. =20

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 10/1/2004 7:45:40 AM
Subject: Re: Stage diameter, Spill strip clearance, thermocouple temp

Hi Phong,

Thank you for the gland diameter data - I've also found the spill strip clearances.

With regard to temperatures, what we really need is the measured temperature at the DE and GE wheel during the test - this will enable us to see how the wheelcase temperatures are affected by cooling steam flow.

Ideally, I believe we should try to obtain more data from Unit 1 because we see a significant change in flow between the 8 turns closed and the 10 turns closed positions. The following data would be very useful for SRCV positions at 8, 8-1/2, 9, and 9-1/2 turns closed:

Pressure downstream of FMO
Temperature downstream of FMO
Steady state temperatures at DE and GE wheel case positions for each valve position

This data will allow us to predict the P5 at which the TE steam flow becomes swamped by (postulated) diaphragm leakage.

Please note the following:

I have in my possession a table dated 5/2/03 entitled "IPSC's Steam Turbine HP to IP Rotor Cooling Steam Data". The implication from this is that we had 12372 lb/hr cooling steam flow at 828 deg F prior to retrofit. The proposed system is perfectly capable of handling this flow at somewhat lower temperatures and I feel, therefore, that there is no requirement to achieve a 9 psi pressure drop across the central gland. In light of this, you may wish not to reconsider the design parameters for the augmentation proposal.

Best regards,

Alan

:.________________

CC: <robert.cunningham@power.alstom.com>

From: Bill Morgan
To: Phong Do; Richard Schmit
Date: 9/2/2004 12:55:50 PM
Subject: Re: Your inputs

Phong,

See attached word document.

>>> Phong Do 09/01/2004 10:58:50 AM >>>
Bill, Richard,
Please address Alan's (Alstrom) questions below and email your comments/answers. Please refer to drawing B&V 9255.62.1001.05-10200. Thanks.

"Phong, One thing I am not clear about is your standard mode of operation - your guidance would be useful.

I assume that for cold starts HSPV is used to pressurise the HP casing and warm it through by condensation heating. This would normally take place while condenser vacuum is being raised, at which point the IP rotor will also be heated through HSFV from the gland packing lines.

Once the turbine is ready to roll, HSPV will close and the HP cylinder will depressurise.

HSFV will then close and S11 will open.

If I am correct on the above assumption then I foresee no problem. If, however, S11 is programmed to open before the HP cylinder depressurises then there may be a drainage problem and we may need to consider a drain orifice to bypass the new pneumatic isolating valve.

Note also, do you believe it would be necessary to supply this valve with limit switches? These could be arranged in parallel to the limit switches on S11 to signify correct operation.

An estimate of the performance benefit and the mark-up to 903E892 will follow.

Regards, Alan"

Turbine Warming for Startup

Warm with Auxiliary Steam through the HSPV (MBV-16), which supplies aux steam into the Cold Reheat line back to the HP.

The operators have sometimes opened the HSFV (ACV-21), by selecting rotor-shell warming on the EHC panel with the turbine tripped, which allows a little steam flow into the IP section.

When main steam temperature and pressure has been raised to about 400 deg F and 250 psi, the operators will close the HSPV and reset the turbine to use pre-warming through the Main Valve #2 Bypass. In this mode, the Control Valves are 100% open and the Stop Valves are 100% closed. Steam is admitted through the Main Stop Valve Bypass in a controlled operation to warm the HP section.

When the rotor-shell has been warmed and the Control Valve Chest requires warming then this is also accomplished through the pre-warming mode selecting chest warming. Pre-warming is selected to Off, which closes the valves, and then Chest Warming is selected. Steam is admitted through the Main Stop Valve #2 Bypass in a controlled operation until the control valve chest is warmed.

Once warming is completed the pre-warming mode is selected off, this shuts off the Main Stop Valve #2 Bypass. This also shuts off the HSFV and opens the Reheat Rotor Cooling Valve (ACV-19)

The operator then prepares for turbine roll-up and selects speed of 200 RPM.

From: Richard Schmit
To: Bill Morgan; Phong Do
Date: 9/2/2004 1:28:08 PM
Subject: Re: Your inputs

Bills sheet looks accurate. I think the limit switch indication would also be a good plan.

Richard F. Schmit
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IPSC Operations
435-864-6016

From: <alan.lang@power.alstom.com>
To: Phong Do <PHONG-D@ipsc.com>
Date: 9/30/2004 5:36:40 AM
Subject: Re: Question on the Report

Hi Phong,

The pressure of 454 psi is what I expect to see at the temperature measuring points. If you use the data taken by Mike Knibb (15/04/2003) and plot flow versus P4 (pressure downstream of the FMO) you will also see that flow asymptotically approches zero at a pressure of 454 psi. I believe we can be very confident with this estimate.

Note also that the pressure drop measured across the FMO is very much greater than the pressure drop across the stage 8 diaphragm glands, so the FMO is actually the flow limiting factor in this part of the system.

I don't have full information to hand, but if we assume a rotor diameter of 36" and five 'see through' gland fins of 0.035" radial clearance then a rudimentary gland flow calculation gives us a flow of 13800 lb/hr with a 9 psi pressure drop. Since there are two such flow paths (TE and GE) then we can see that the central gland area does not offer very much resistance to flow.

Hope you agree with this analysis.

Best regards,

Alan

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CC: Dave Spence <DAVE-S@ipsc.com>, <robert.brown@power.alstom.com>, <robert.cunningham@power.alstom.com>

From: Phong Do
To: alan.lang@power.alstom .com
Date: 9/29/2004 11:33:57 AM
Subject: Question on the Report

Gens,
Thank you much for helping us to understand the report and proposal better. I would like to know:

1. Is P5 = 454.3 psi = The calculated pressure for the IP double flow at the thermocouple areas (GE's DWG 148D5583)? The drawing shows P5 design = 442.3 psi. If P4 = 457.5 psi, there is only 3.2 (457.5 - 454.3) psig dp across the gland. Is this dp enough to force the IP steam across the gland and enough to cool the wheel, thermocouples...
2. What is the current P4 value?

Thanks.

CC: Dave Spence; robert.brown@power.alstom.com; robert.cunningham@power.alstom.com

IP7008502

From: Phong Do
To: alan.lang@power.alstom.com
Date: 8/23/2004 7:56:52 AM
Subject: Re: New S11 Valve for the New Line

Thanks, Alan, Go ahead to use and spec out a parallel slide valve for the new S11 and let's use a globe valve for manual throttling.

>>> <alan.lang@power.alstom.com> 8/23/2004 7:22:19 AM >>>
Hello Phong,

Re: parallel slide valves

We have extensive experience of using parallel slide valves in steam systems - they are are available at standard and special ANSI ratings up to 2500 lb Special class in all common materials, e.g carbon steel, 1-1/4 CrMo, 2-1/4 CrMo. Seat materials are stellite faced.

The valve has several advantages over the globe style valve:
   Reduced pressure loss due to see through design.
   Reduced actuating loads since the valve is effectively pressure balanced.
   The valve seat is out of the flow region and is not subject to erosion in service.

The only disadvantage, that I can think of, is that the parallel slide valve is not suitable for throttling duty - the flow is concentrated at one point on the valve seat rather than dispersed around the circumference as in a globe valve. This is not particularly an issue since we are not proposing to throttle with the valve.

Regards,

Alan

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